NEW YORK SEATTLE

MOTORSHIP

Devoted to Commercial and Naval Motor Craft

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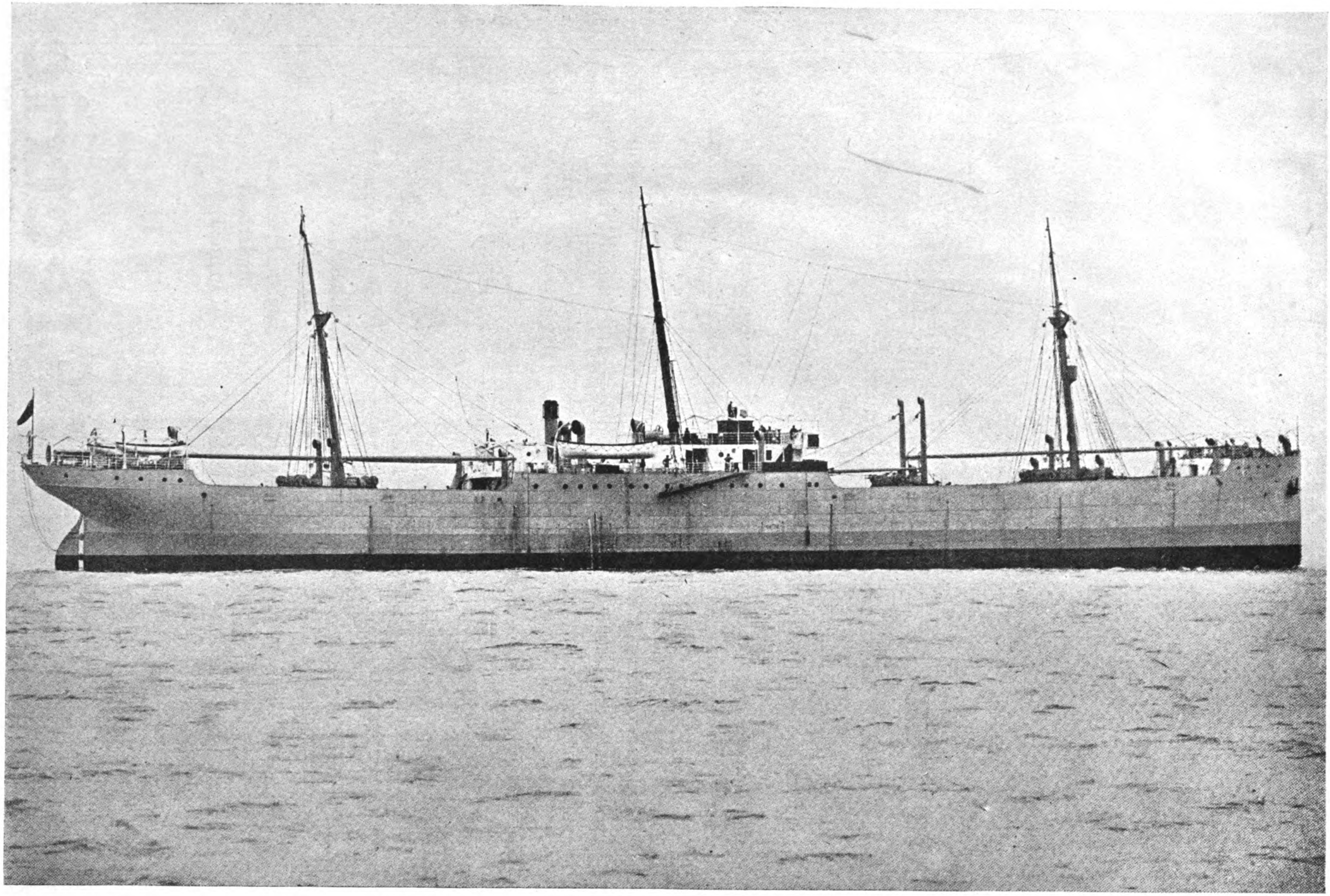

NOTEWORTHY ECONOMICAL MERCHANT SHIPS—No. 33

Twin screw steel motorship "Kangaroo." owned and operated by the Commonwealth of Western Australia. This ship was built in 1915 by Harland and Wolff, Ltd., and the engines installed by the same firm were supplied from the Copenhagen shops of Burmeister and Wain. The vessel is 365 ft. 0 in. in length between perps, 50 ft. 1 in. in breadth, and 26 ft. 6 in. in depth. Her deadweight capacity is about 6,600 tons, whereas a similar sized steamship would have a d.w.c. of only about 6,200 tons. Her engines are twin B. & W. standard single acting 4 cycle, 6 cylinder units, the cylinder dimensions of which are 22 in. x 29 15/16 in. and, developing 1,125 i.h.p. each, at 140 r.p.m. She carries 874 tons of bunker fuel, has complete electrically driven auxiliary for the engine room and deck, and electro hydraulic steering gear supplied by 2—100 K. W. Diesel-electric generators. Total consumption of fuel oil for the main engines and all auxiliaries amounted to only 0.292 lbs. per i.h.p. per hour on her trials. Her sustained speed is 11 knots.

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MOTORSHIP

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PUBLISHED MONTHLY IN THE INTERESTS OF COMMERCIAL AND NAVAL MOTOR VESSELS AND FOR RECORDING PROGRESS OF THE MARINE INTERNAL-COMBUSTION-ENGINE

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Great Britain. W. S. Smith & Son. Ltd., London. Australia. Gordon & Gotch, Sydney & Melbourne.

The oil-engined motorship has arrived! It is such a pronounced economy that it was bound to come. Nothing could stop it! And all obstacles have been removed as fast as they arose. The law of progress has seen to that. Very strong prejudices stood in the way of steam. But, one after another they were swept aside and steam reigned triumphant for a century. Steam now has had its day! Its zenith has passed, and gradually but surely it is being superseded by the economical internal-combustion power. Steamships are becoming decadent. America, the most important oil-producing country, is to be the greatest motorship-owning nation. Let us all co-operate and assist to make that day soon.

October, 1919 Vol. 4 No. 10

EDITORIAL

A GREAT AMERICAN MOTORSHIPPING ENTERPRISE

Newcastle-on-Tyne, 1st Septr. 1919.

IN the United States important affairs generally are arranged and completed far more swiftly than in the Old World, due to the natural "pep" and rapid-fire decision forming methods of her big business men, of which many are of British extract or blood vitalized by American spirit and system; although, this remark is not intended to belittle the great things frequently accomplished in Great Britain and Europe. And, partly because of this condition the words in italics at the top of this page show every sign of being completely realized within a very few years. Yes! they are nearer being an accomplished fact than ever before.

Traveling through the leading shipbuilding districts of Scotland and England we find that, with about a dozen exceptions, the shipowning, shipbuilding, and marine engineering people are inclined to smile at, rather than take seriously, the friendly maritime-trade competition that is bound to arise in the near future; evidently basing their judgment upon our ship construction as carried out during the war period by some of the new yards under Federal control, where most of the head officials had never even seen a ship before and where clergymen, actors and bootblacks were employed as riveters, etc. Also, because prior to the war our shipping efforts were far too feeble to seriously endanger Great Britain's mercantile supremacy.

Little do they realize the grim determination of our big financial interests to carry on shipbuilding and ship operating in a manner never before attempted with or without the Government's assistance, and to settle down to the task before them in a thorough and serious manner. Like the phoenix ascending from the ashes, so will our merchant fleet successfully rise from the embers of our blunders.

Re-organization from war-work is being completed much more quickly in the United States than it is in Great Britain, the latter country having had five years of war compared with our two, also generally speaking they are slower in decision and action, and their labor troubles are even worse than our own. Furthermore, the British Admiralty has only just canceled all naval work, and many naval craft are still on the ways partly completed and have to be removed or completed before they can be replaced by the keels of merchant ships. Lastly, many Britishers have not yet learned the true value of keeping their name prominently before the public while under the process of re-organization for peace production. Meanwhile a number of American companies are securing world-wide publicity and energetically seeking business abroad to handle when their present capacities slacken. No wonder the exchange rate is so favorable to America! Great Britain badly needs foreign orders for her products in order to improve her credit and restore to its former value the English pound sterling.

America's new ships—for undoubtedly motorships will be built—will be numbered among the most modern afloat, and will be equipped with the most rapid cargo-handling devices. A change of opinion is coming over the more conservative of our shipowners who now are beginning to realize the great economic advantages of oil-burning motorships.

One of the strongest indications of the realization of those words in italics, which we have printed for over a year, is the recent consolidation of the Kerr Navigation Company of New York, and the Wm. Cramp & Sons Ship & Engine Company of Philadelphia. This powerful combination means the greatest impetus ever given to the motorship movement in America—and, probably the greatest in the world. This new concern—the American Shipping & Commerce Corporation—has a capital of forty-six million dollars ($46,000,000.00) and is backed by such strong financial interests as Hayden, Stone & Co., and Chandler Bros. It incorporates one of the oldest and largest shipbuilding concerns in America with a progressive steamship-owning company who have made millions of dollars by successful ship-operating during the last few years. What is more important, so far as we are concerned, is that both companies are

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united in their strong belief in the Diesel engine for cargo and passenger ship propulsion; and, incidentally, we can modestly say that we had not a little to do with their present knowledge and appreciation of what motorships mean to our country's mercantile marine. The Cramp shipyard, of course, controls the construction in America of one of the most successful of European marine Diesel engines, and undoubtedly they will make every effort to develop it on a large scale. So we can look forward to seeing a big fleet of splendid, fast, and modern motorships.

We have been advised personally by important officials of the Kerr Company that they will not consider the construction of steam-driven vessels, and that they are convinced that motorships are the only thing. Unfortunately, such a fleet cannot be placed in service by waving a wand, but will take some years to complete. However, we must remember that it has taken the well-known East Asiatic Company over nine years to secure its fleet of about fifteen fine motorships. Nevertheless, we can feel assured that it will not take the American Shipping & Commerce Company so many years as that, and their vessels also will trade in the East. We look forward to their motorships being ahead of anything yet afloat in the way of economy, speed and capacity.

Doubtless their splendid lead will be followed by other great domestic organizations such as the American International Corporation, and its several shipowning and shipbuilding associations, which include the Pacific Mail Steamship Co.; W. R. Grace & Co., and the New York Shipbuilding Corporation, the latter concern being the holder of a license of another leading European Diesel engine, which they are just about to start building.

Official Test Report De La Vergne Heavy-Oil Marine Engine

(Continued from September Issue)

14 DAY DURATION TEST

Attendance.

The crew on the engine during the Duration Test of 14 days was divided into three watches.

Watch	Time	Foremen
1.	12 Midnight to 8-A.M.	Mr. Schlumberger
2.	8-A.M. to 4-P.M.	Mr. Ulderup
3.	4-P.M. to 12 Midnight	Mr. Banner

The foremen were responsible for the engine and for the tests made in their time.

Duration.

The test started on March 23rd at 1.05 A. M. The engine was kept running until April 6th to 2.05 A. M. The duration of the test was exactly two weeks (one hour difference on account of advanced clocks).

The load was absorbed by a water brake designed by the De La Vergne Co. The water passing through the brake was controlled by hand by the operator. During night time when the water pressure was constant in the pipe lines the valves were adjusted once and the load stood constant during the whole night. In day time when water from the line was used for different purposes at other places the water pressure changed quite frequently and the man had to regulate the quantity of water supplied to the water brake.

Speed.

The speed was kept at 200 R.P.M. It was constant during the night and changed somewhat during the day according to the load or the water pressure in the brake.

Repairs.

The following repairs were made while the engine was under test.

March 25th. The bearing between flywheel and horse shoe of thrust bearing ran warm.

March 27th. Thrust shoes arranged for water cooling. Pipe connection made.

March 28th The small overflow valve on the fuel pump No. 4 broke and the pin operating it bent. It was replaced without changing speed or load on the engine. The pumps of the other cylinders were taken out one after the other and the overflow valves and stems examined.

March 30th. One of the four cylinder lubricating pipes on cylinder No. 3 broke. The opening was plugged up with a wooden plug and the cylinder lubricated with three pipes only.
The roller for the fuel pump No. 3 stuck on account of salt water having been splashed into the oil cup from the broken cylinder lubricating pipe. This roller was not replaced and the test finished with the roller not turning on the pin.

March 31st. Connecting rod for McCord lubricator broke. Replaced by new one. Slowed down to 150 R.P.M. for about three minutes.

April 1st. Suction pipe of fuel pump No. 2 blown out with air in order to clean it. Cylinder No. 2 was not working for that purpose from 4.15 to 4.40. Nothing else happened until the end of the test.

Exhaust.

The exhaust seen at the exhaust pipe outside of the Test Room was slightly gray during the whole test. No color could be seen usually on the small test flanges which are on the exhaust elbow of every cylinder. The temperature of the exhaust was measured at the exhaust elbow for every cylinder with a pyrometer. Care was taken to have the temperature of the exhaust from all four cylinders alike as much as possible. It was between 580 and 670° F. during the whole test.

Lubrication Oil Consumption.

The amount of lubrication oil given in the engine including thrust bearing was measured several times. The results of these measurements are shown on the following page.

The total amount of oil supplied to the engine was .067 lbs. per B.H.P. hour. All this oil was circulated and fed to the engine again. The total loss of oil was 0.0078 lbs. per B.H.P. hour when no oil reclaimer was used. If the oil is reclaimed the total loss will be but 0.0008 lbs. per B.H.P. hour.

Fuel Consumption.

The fuel consumption of the engine was measured continuously. Each watch measured the oil at the start, the amount of oil put in the tanks

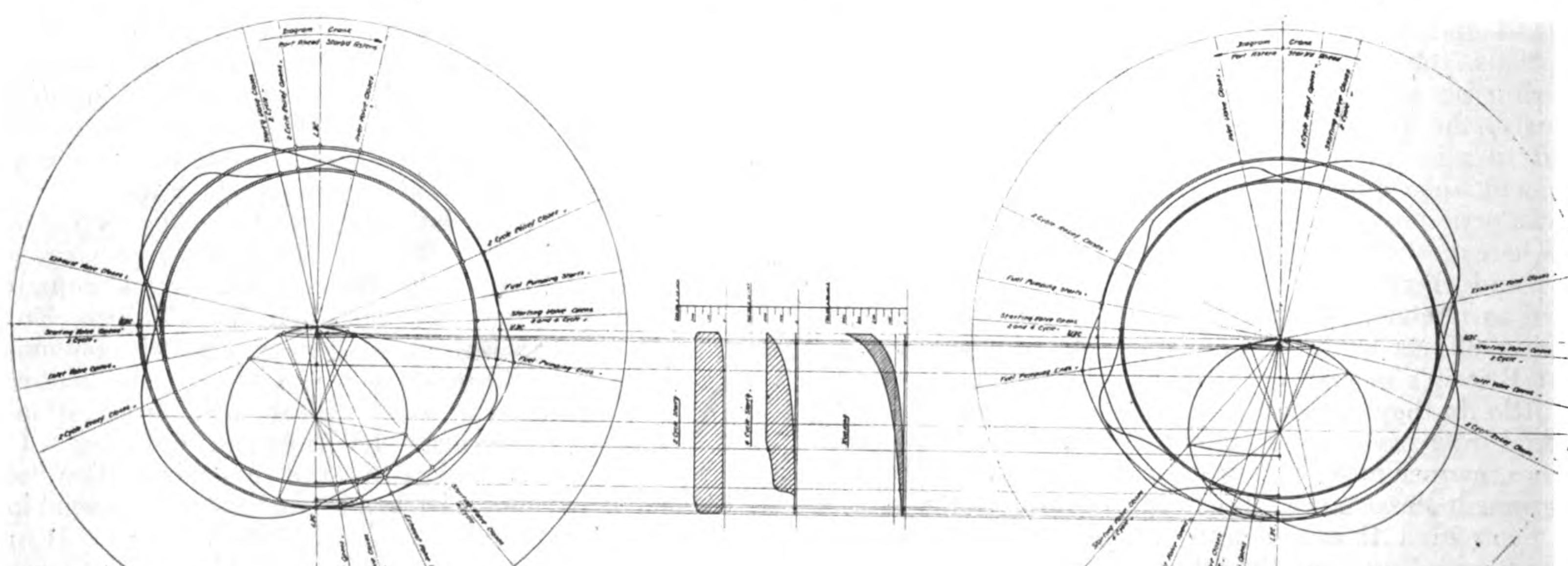

Fig. 7

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Date	Shift	Hours	Taken from Storage Tank for: Cylinders	Wrist Pins Pump roller Pins Hand Oiling	Main Bearings Crank Gears Thrust Bearing	TOTAL	Recovered	Loss	TOTAL
April			lbs. oz.	lbs. oz.	lbs. oz.	lbs. oz.	lbs. oz.	lbs. oz.	lbs. oz.
4	8— 4	8	19— 8	28— 2	149—	....	175—14	20—12	....
4	4—12	8	28—	13—	163— 8	....	181—	23— 8	....
5	12— 8	8	14— 6	19— 8	210— 2	....	220—	24—	....
5	8— 4	8	17—12	27—	153—	....	180—	17—12	....
Totals		32	79—10	87—10	675—10	842—14	756—14	86—	842—14
lbs. p. B.H.P. per hour			.0062	.0068	.0527	.066	.0592	.0067	....
Gal. p. 1000 B.H.P. p. hr.			.825	.905	.703	8.8	7.88	.900	....

Consumption of Lubricating Oil Running at Full Load

during the watch and the amount of oil left in the tanks when leaving. The fuel consumption given by each watch is therefore the average of 8 hours. During the whole test it averaged 0.41 to 0.42 lbs. per B.H.P. hour.

Cooling Water.

The cooling water was delivered by a water pump attached to the engine. The engine is equipped with two pumps for this purpose. During the tests only one pump was used and it supplied more water than was necessary. The average inlet temperature of the water was 40 degrees F. The average outlet temperature measured on the outlets of the cylinders was 69.4° F.

Temperature in the Crank Case.

The temperatures in the columns near the main bearings were measured towards the end of the duration test. At the same time the temperature in the room about 8 ft. from the engine was measured. The results of these measurements are shown on table No. 2. They show that the temperature inside of the engine was only 33.87° F. above the room temperature.

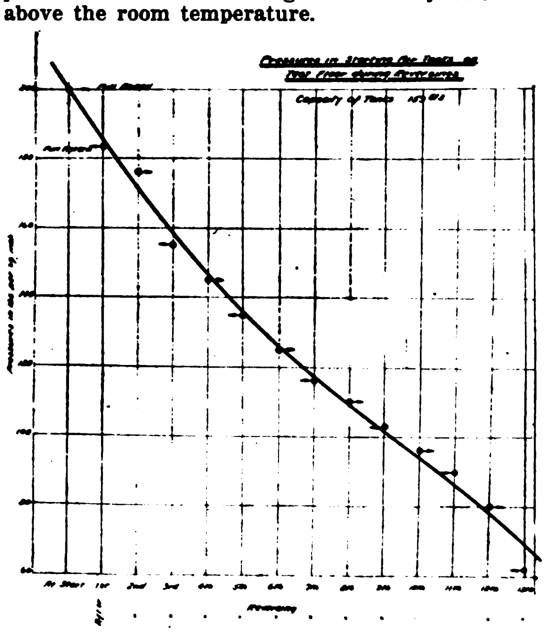


Fig. 8

Running with Different Loads and Speeds.

To determine the flexibility of the engine, tests were made to run the engine from 50 b.h.p. to 500 b.h.p., with a speed of 75 to 200 r.p.m. The results of the tests in connection with the fuel consumption is shown in figures 9 and 11. Both figures show the exact range of speed and power within which the engine will run to any length of time without any difficulty.

Heat Balance

The heat balances of the engine on the different loads and speeds are shown in fig. No. 10. The friction of the engine was figured as 12% of the indicated horsepower. This figure was taken from another engine in which exact tests were made and the mechanical efficiency was determined to be between .88 and .90 at full load.

Starting.

In order to start the engine in any position, starting air is admitted in two stroke cycle first and after it is moving, in four stroke cycle. No oil is admitted during two stroke cycle starting. In four stroke cycle starting air and oil are ad-

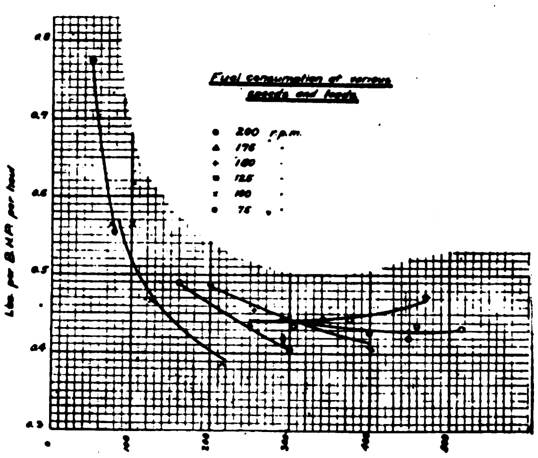


Fig. 9

mitted. The oil is ignited at the start by a coil which is heated by electric current, delivered from a storage battery. The heating coils operate on four volts and 120 amperes. They are connected in parallel. After the first ignition occurs the starting air is turned off and the engine runs on fuel oil. Then the current for the coils is switched off.

Reversing.

The reversing of the engine consists of turning off the oil and bringing the engine to a stop. After that the engine is started in the opposite direction. The starting coils need not be turned on for reversing. The oil ignites without it. Reversing is accomplished from full load ahead to full load astern in 10 to 12 seconds. The pressure of the starting air in the tank is 200 pounds per sq. in. maximum. The engine was reversed on the test floor with as low a pressure as 62 lbs. per sq. in. The starting air tanks on the test floor had a capacity of 159 cu. ft. The tanks filled up with air at 200 lbs. per sq. in. are sufficient for 13 re-

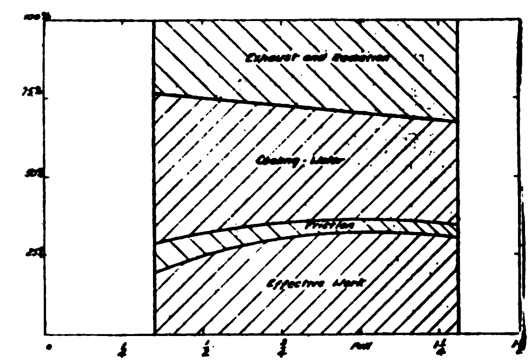


Fig. 10. Heat Balance

versals. The drop of the pressure in the tanks in reversing is shown in figure 8. The tanks on board ship will have a capacity of 200 cu. ft. for each engine. The rules of the American Bureau of Shipping call for tanks with sufficient capacity for 10 reversals without refilling.

To show the ease and fool proofness of the reversing gear the engine was reversed about 200 times by employees of the De La Vergne Machine Company.

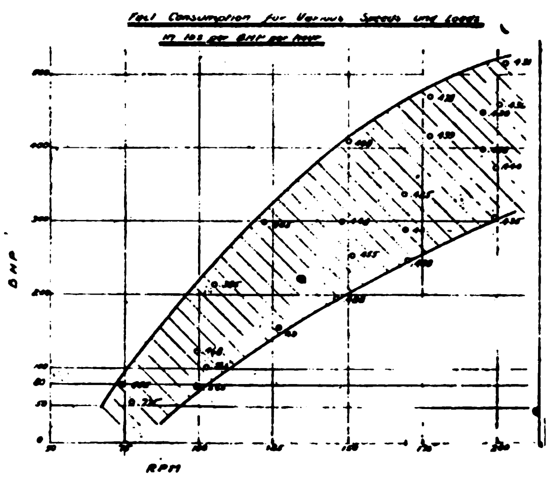


Fig. 11

Fuel Oil Analysis.

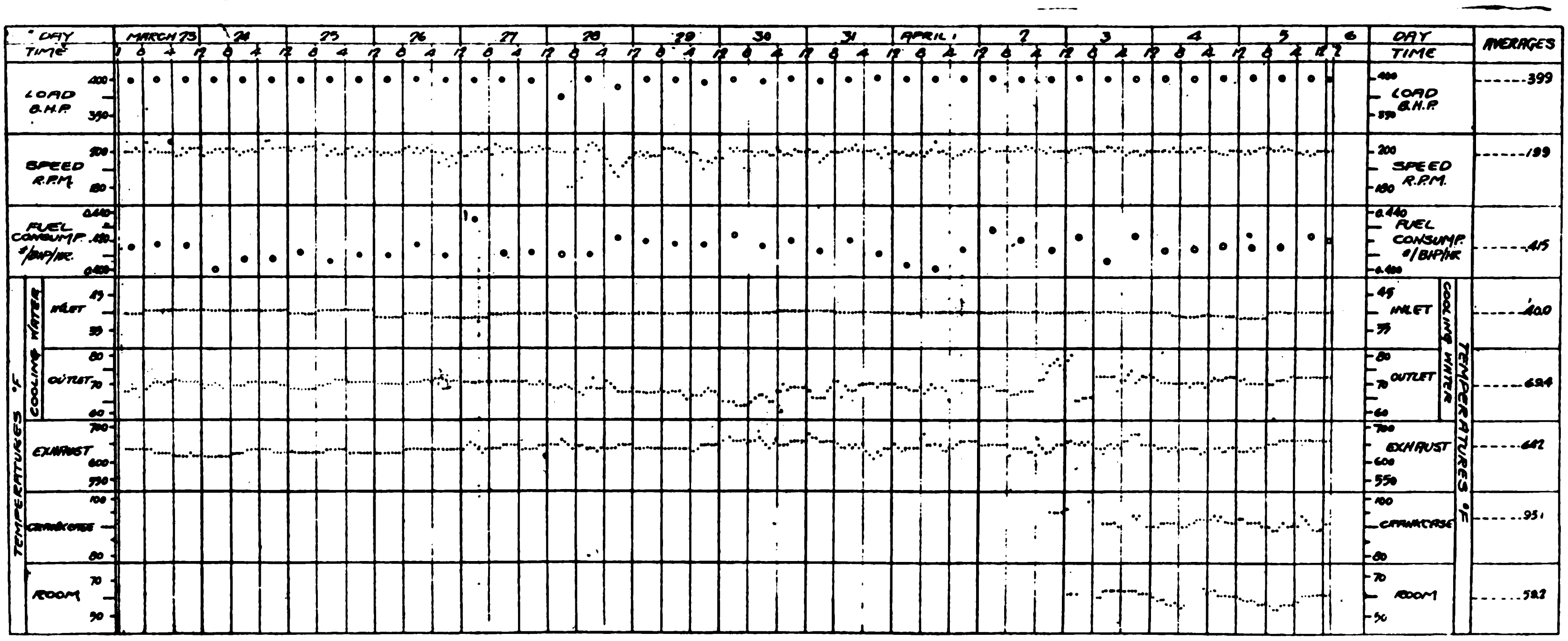
The fuel oils used during the tests were of two grades, a light and a heavier oil. The heavier one was used for about two or three days. The lighter was used during the rest of the tests.

Light Oil

Specific gravity	.8674
Saybolt viscosity at 100° F.	49
Flash Cleveland Open Cup	195°F
Fire Cleveland Open Cup	215°F
Tar (Asphalt)	.786%
Moisture	.6
Ash	.105%
B. T. U.'s	18,795;18,744

Heavy Oil

Specific gravity	.8794
Saybolt viscosity at 100° F.	67
Flash Cleveland Open Cup	200°F
Fire Cleveland Open Cup	230°F
Tar (Asphalt)	.542%
Moisture	1.0%
Ash	.16
B. T. U.'s	18,677;18,712



Test Chart of Two Weeks' Non-Stop Run

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Marine Thrust Bearings

Remarkable Results Obtained from Applications of New Theories

An inherent defect in a design or in the application of a principle may not become manifest under ordinary conditions. But when conditions become abnormal a correct application of fundamental laws and greater care with details will generally solve the problem.

One of the greatest recent improvements in engine room accessories is in the design and construction of the main thrust bearing which takes the full thrust of the propeller. At this point all our efforts for added power output are made manifest in the increased propulsive thrust of the screws

Before the advent of the direct-connected turbine, the thrust block received a large amount of careful thought. With the introduction of the marine turbine, the power transmitted through one shaft became much larger. The thrust block did not, however, enter into any of the installations, as the axial component of the steam against the turbine blading was of itself the proper reaction to counteract the thrust from the propeller.

With the adoption of reduction gears between the turbine shafting and the propeller, the problem of maintaining the fore and aft position of the shaft again became an important one. The old type of multi-collar horseshoe thrust was designed for the increased powers and little thought was given to its possible failure. It became evident very quickly, however, that the old type of thrust bearing had reached the limit of its usefulness.

Fortunately, it is seldom that theory and laboratory knowledge are not well in advance of actual practice. This was especially so in the case of thrust bearing design.

The theories of Osborne Reynolds, which he discussed in a paper read before the Royal Society of Arts in London in 1886, showed that perfect lubrication consisted of interposing a wedge-shaped film of oil between two rubbing surfaces so that there would be no metal to metal contact. It was demonstrated that a journal revolving in a bearing has a slight side play and this is illustraed in Fig. 1. The ability of the oil to cling to the surface of the journal and to be swept into the resulting wedge-shaped space has been known for a long time. That phenomenon underlies the successful operation of all shaft bearings, although as yet its importance is not recognized as widely as it should be.

In the discussion that closely followed the presentation of Professor Reynolds' paper the suggestion was made that film lubrication could be maintained in a thrust bearing if one of the bearing surfaces were formed with shoulders symmetrically spaced around the thrust collar; the face of each shoulder to be slightly bevelled in the direction of rotation (Fig. 3a) to each radial oil groove, in the direction of rotation. This same idea has occurred to many minds, but it remained for two eminent engineers in widely separated parts of the world to appreciate the full significance of Professor Reynolds' work. One of these was Professor Albert Kingsbury of the United States of America, who has become widely known during the past thirty years because of the merit of his original researches in the field of lubrication. The other was Mr. A. G. M. Michell, a brilliant English engineer residing in Australia.

The thrust bearings, invented independently by these two engineers, are identical in their simplest form, which consists of a plain thrust collar bearing against pivoted segments. This type of thrust bearing was patented in England by Mr. Michell and in the United States by Mr. Kingsbury. It is known by the names of both inventors or by the descriptive name of *pivoted segment type* thrust bearing. Its development has been rapid in the past five years.

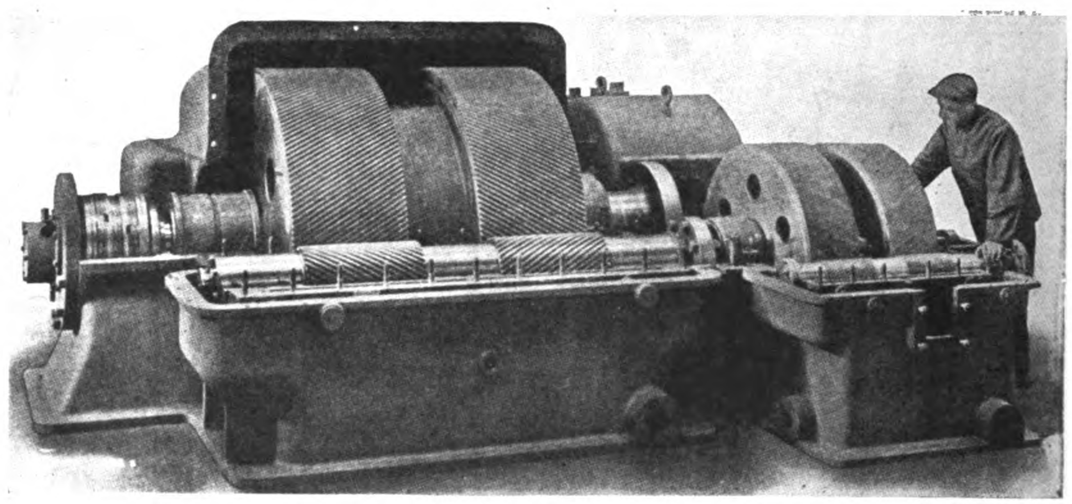
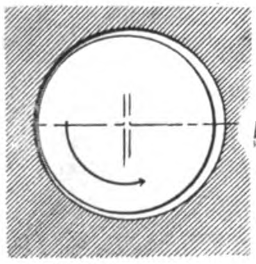


Plate A.



"Engineering"
Fig. 1.

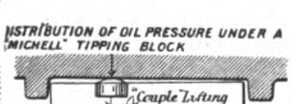


Fig. 2.

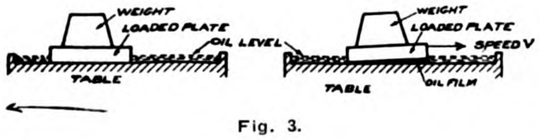
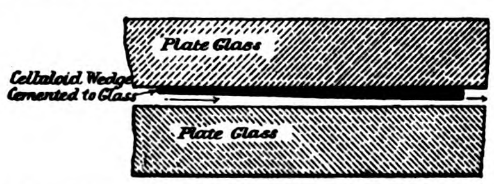


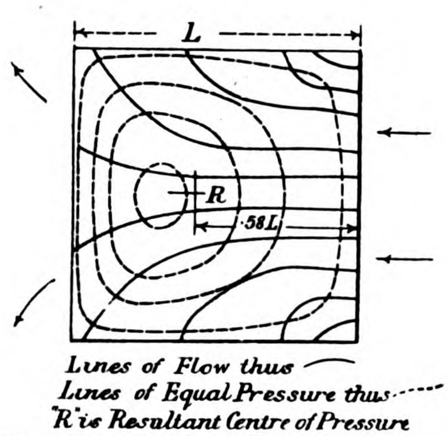
Fig. 3.

The Michell bearing developed first in the Marine field while the Kingsbury bearing developed in the stationary hydro electric field. Both developed simultaneously in the steam turbine field. The Kingsbury thrust was first used in the Westinghouse turbines and the Michell thrust in Parsons turbines.

The pivoting of the segments permits them to accommodate themselves to the different shapes of oil films required by different speeds, loads, and oils. The proportions of the oil wedge vary with all those elements. Hence it is this freedom of movement of the elements of one of the bearing surfaces that is the fundamental and eminently important invention of Professor Kingsbury and of Mr. Michell.



"Engineering" Fig. 4 (A)



"Engineering" Fig. 4. (B)

The phenomena of film lubrication is illustrated in Fig. (2) and (3) and may be described as follows: A plane surface upon which rests a block with a weight on it is covered with oil. If this weighted block is given a velocity V by sliding it along the plane surface, the leading edge will be seen to rise and a film of oil will be interposed between the moving block and the table.

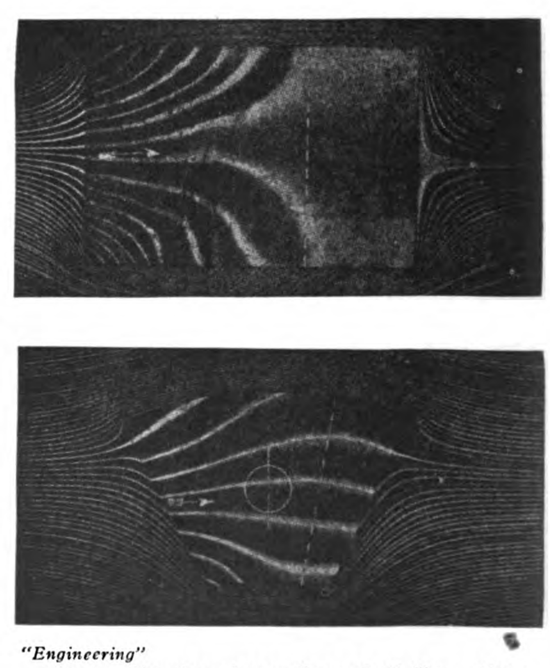
This film will vary in thickness and a study of its action is illustrated by Figs. (4) and (5). It is found that the pressure of the film under the surface of the moving block varies and the stream lines of the oil composing the film show the tendency to spread which is also evidenced by the loss of oil at the sides.

With blocks that are too long or wrongly pivoted the film at the following edge becomes too thin and finally breaks down allowing metallic contact and friction. The results of experience indicate that the blocks should be about square and pivoted at a point approximately 60% of their length from the leading edge. It is common and successful practice, however, to use blocks centrally supported.

The film thickness varies with the rubbing speed and pressure. Since the load is floated on the oil it is evident that the sum of the pressures within the liquid must equal the total load. Thus the mean pressure is known, but the actual pressure varies from nothing at the edges of the plates or pads (see Fig. 4) to a maximum about under the centre of pressure. This point is about 60% of the length of the plate from the leading edge and the actual pressure is about twice the mean pressure. The friction of the pads when floating on the oil film depends upon the surface speed, viscosity of the oil and the film thickness. The first factor is under control, as is the second as far as the temperature of the lubricant will affect viscosity. The film thickness will vary according to conditions. This thickness at the leading edge will be about double that at the trailing edge and its mean value may be as much as a thirty-second of an inch for a light load and high speed with heavy oil. With a heavy load at slow speed the oil film may decrease to less than a thousandth of an inch.

There is a relation between the points of support where the maximum load may be carried and where a given load can be carried with the least friction. These two points are nearly coincident and it would seem at first glance that they should actually be so. With this type of bearing wherein the oil film supports all the load it would seem that less friction would mean less resistance to internal cleavage. Evidently the oil film should be able to stand the greater load when located at that point.

The relative coefficients of friction in thrust collars are as follows for similar service conditions:



"Engineering"
Fig. 5. Showing stream lines of oil film. Note that the long rectangular block loses the film at the trailing edge

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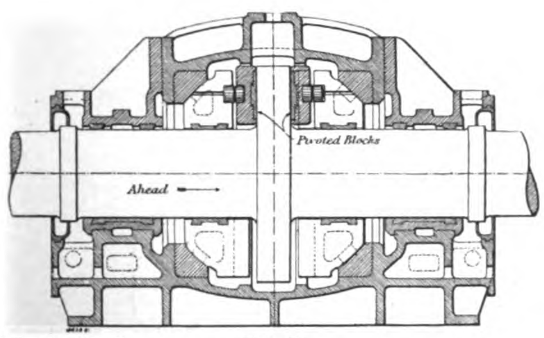


"Engineering" Fig. 7. Section of "Michell" thrust

Ror Rocking Pads	0.0015
For Rigid Collars	0.0300

The old type giving 20 times more friction.

In an experiment conducted by the Westinghouse Co. a pressure of 7,000 lbs. per sq. in. was carried at a rubbing speed of 54 ft. per second. The coefficient of friction being 0.0008 to 0.0003.

To show the effect of shifting the position of the pivoting point in bearings of this type the temperatures of the oil have been taken after running with the pads supported at different points.

Location of support	Temperature of oil
1-16 inch forward of centre	135°
Centre	132°
1-16 inch behind centre	126°
⅛-inch behind centre	124°

The coefficient of friction varied from 0.002 to 0.001.

It must be borne in mind that the presence of the oil film between the bearing surface is due solely to its viscosity and a surprising resistance against being wiped off the collar by the pads. When starting up thrust bearings while under load there is metal to metal friction to a certain degree and the bearing groans for a fraction of a revolution. The oil film (see figs. 4, 5 and 6) has by that time become thick enough to separate the surfaces. It thickens as the speed increases.

The general rules that have become well established for bearing practice concerning the bearing metals and oils used, apply to the new type as well as to the old one.

For low speeds and heavy loads a thick oil must be used and brass shoes against hardened steel collars will give no trouble. For high speeds and light loads a thin oil is preferable and the shoes should be faced with babbitt metal, the collar being cast iron or mild steel.

The coefficient of friction is found to vary directly as the square root of the load and inversely as the square root of the unit pressure. This is the rule established by Reynolds in his treatise referred to above. An approximate rule for a bearing loaded to about 350 lbs. per sq. in. with the oil kept at 40° C is 0.00009 x V x r.p.m.

For marine work thrust bearings of this type are usually designed for bearings pressures of about 200 to 300 lbs. per sq. in. instead of the 20 to 80 lbs. allowed on the rigid horseshoe collars. This enables the thrust shaft to be made with only one collar instead of 10 or 15. The impossible task of keeping such a number of collars all adjusted to carry equal pressures is evident. A slight increase in temperature is sufficient to spoil such an adjustment and increased friction and more heat loss is the immediate result. The latest horseshoe thrust bearings are always watercooled.

Discussing briefly the design of journals revolving in bearings it has been known for some time that an approximate film lubrication was secured. The great discrepancy between the unit pressures permitted in such bearings and those obtained with perfect film lubrication shows conclusively that the film is not maintained. The practice of the General Electric Company in designing journal sizes has been as follows:

Rubbing Speed Feet per min.	Pressures Allowed per sq. in. of projected area
20	168
30	190
40	208
60	229
73-½	235

This we find does not compare favorably with the strength of an oil film. In the newer types of thrust bearings pressures of over 9000 lbs. per sq. in. have been carried at a rubbing speed of 50 ft. per second without breaking down the film. The above company's rule for the heat loss in a journal is

$$\text{B.T.U. loss per hour} = \frac{190\ \text{l.d.v.}}{t - 32}$$

where

l = length of journal in inches
d = dia. of journal in inches
v = rubbing speed ft. per sec.
t = temperature in degrees Fahr.

For purposes of comparison it may be of interest to give a few empirical formulae used in the design of the old style horseshoe collar thrust block. These were obtained from a paper on the "Design of an Ideal Thrust Block" read before the Institute of Marine Engineers by C. P. Tanner. From fundamental principals the thrust in pounds is equal to

$$\frac{\text{I.H.P.}}{S} \times Y \times 33{,}000$$

where S = speed of advance of the ship in feet per min.
Y = propulsive coefficient

The value of Y is the product of all the efficiency factors entering into the production of the propulsive thrust.

These various efficiences are as follows:

em = mechanical eff. of the engine.
ep = efficiency of the propeller.
eh = efficiency of the hull.

It is to be remembered that the propeller works in disturbed water that has a slight forward motion. This is called the wake gain and the true speed of advance of the propeller is generally about 90% of the apparent speed of the ship.

Bearing surface required for the collars in square inches.

$$S = 0.6 \times \text{I.H.P.}$$

Allowable pressures on the bearings are

50 — 80 lbs. per sq. in. Merchant
80 — 100 lbs. per sq. in. Naval

The number of collars should be approximately

$$N = 1 + \frac{d - 5}{c}$$

where d = dia. of shaft in inches
c = 1.8 for merchant work
1.25 for naval work

Thickness of the collars may be taken as

$$T = 0.4\ (D - d)$$

Space between the collars

0.4 (D — d) for solid brass horseshoe bearings.

and

0.75 (d — d) for cast iron shoes lined with white metal.

If the bearing collars are water cooled, etc. the space may be taken as equal to (D—d).

Another formula for the approximate pressure allowable in a bearing is

$$P\ (\text{lbs. per sq. in}) = \frac{800}{\sqrt{Rd + 100}}$$

where

R = r.p.m.
d = dia. of shaft in feet

To calculate the loss of power to be expected in an old type thrust block the formula; loss per cent $= 0.5\sqrt{\text{r.p.m.}}$, will give a fair idea of what to expect.

The coefficient of friction of a Kingsbury or Michell bearing is well under 0.002 on the average and is little affected by the increase in bearing pressure up to extremely high unit loads. The allowable bearing pressure is at least ten times that for the horseshoe thrust.

The saving in weight and space are often vital factors especially the latter when the engine room is placed aft. The cost is also greatly reduced due to the simplifications of the machined parts. The length of a large horseshoe type thrust block may exceed 20 ft. whereas the overall length of a single collar thrust will seldom exceed seven feet. See the comparative plans of the old type and the Kingsbury single collar for the same size of shaft.

In a 30,000 horsepower battleship the power required to overcome the frictional resistance in an ordinary multi-collar thrust block would be 292 h.p. on two shafts. The coefficient of friction being assumed 0.02 which is under the average.

A conservative value for the coefficient in the new types would be 0.002 and thus the frictional horsepower in the same ship would be reduced to 29.2 h.p. on two shafts. This saving of 263 h.p. represents about 420 lbs. of coal an hour at full power. In this instance the entire thrust of 140,000 lbs. on each propeller shaft could be taken on a single collar 32 inch in dia. with a unit pressure of 275 lbs. per sq. in. This is a moderate value for the pressure in the Kingsbury or Michell bearing.

In a recent court action in England the term for the life of Michell's patent was extended for seven years. This was because the war had prevented the general wide spread adoption of the new type of bearing and the inventor would not receive proper remuneration for his inventive ability if his rights were allowed to lapse when due. It was testified that through the decrease in frictional horsepower absorbed by the Michell thrust from that with the old type of thrust block, the Admiralty had saved several hundred thousand pounds during the naval operations of the war.

The result of an exhaustive test conducted by the Westinghouse Co. at Pittsburgh, Pa., on a Kingsbury type of thrust shows conclusively the wonderful persistency of the oil film which is entrained by the revolving collar and prevents metallic contact with the tilting shoe pieces or pads of the thrust bearing.

The particulars of this bearing were

Outside dia. of collar	4¾ inch
Inside dia. of collar	2⅜ inch
Number of pivoted blocks	10
Total area of blocks	10.4 sq. in.
Rev. per min	3470
Mean rubbing speed	50 ft. per sec.

The blocks were steel, faced with white metal 1-16 inch thick and were pivoted behind their centre of area.

With all the blocks in use 1010 lbs. per sq. in. thrust load was carried without temperature rise or any indication of wear.

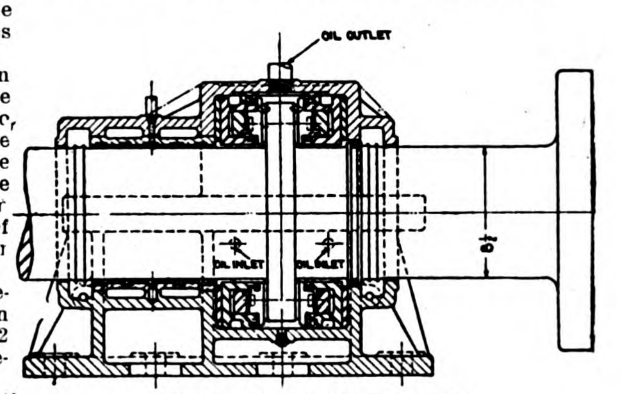
The number of blocks was reduced with the resultant pressure rising proportionately.

No. of blocks	Unit load on oil film
4	2620 lb. per sq. in.
2	5420 lb. per sq. in.

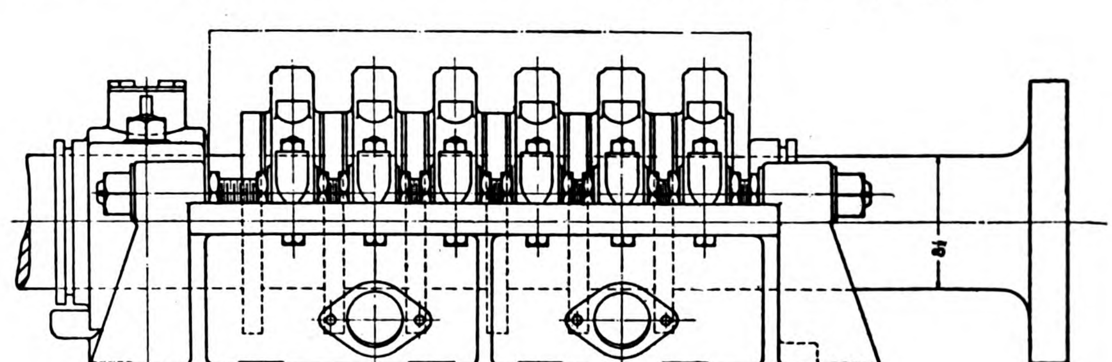
The surface of the two remaining blocks was then reduced to 1.9 sq. in. and the thrust load carried without trouble amounted to 5910 lbs. per sq. in. Then the effective area of the blocks was reduced to a total of 1.2 sq. in. and the load on the oil film became nearly 10,000 lbs. per sq. in. The oil film still remained intact and the bearing did not heat but the white metal squeezed out. This increased the effective area of the bearing surfaces and reduced the unit load to about 7000 lbs. per sq. in.

It is evident from the above results that the limit of load in bearings of this type is the crushing strength of the bearing metal used.

Further particulars of actual marine installations will be of interest. The unit pressures in such cases are of course well under the limits indicated in the Kingsbury experiment as that was a deliberate attempt to load the bearing to destruction.



"Kingsbury" Marine Type. (b)



Old Type Horseshoe Collar Fig. 7. (a)

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The S. S. Cuidad de Buenos Aires and the S. S. Cuidad de Rio de Janeiro were fitted with the Michell bearing.

The vessels were twin screw with geared turbine drive and developed 2625 s.h.p. per shaft.

Diam. of shaft .... 9 inch
r.p.m. 265
Temp. of oil .... 148° F
Temp. of engine room .... 90° F
Pressure per sq. in. of bearing surface 300 lbs.

There was no trouble and no discoloration of the oil after 18½ days' run.

On the U. S. S. Neptune a single thrust collar Kingsbury bearing 13½ inch in dia. carries 4,500 lbs. thrust load at a mean rubbing speed of 72 ft. per sec. and at 500 lbs. pressure per sq. in. A single collar 10 in. dia. bearing on six shoes carries a thrust load of 10,000 lbs. at 350 r.p.m. on each of the twin shafts of the yacht "Whileaway."

A feature of the later Kingsbury thrusts is the provision for equalizing the pressures on all the bearing shoes. This pressure often varies due to distortion of the thrust casing or the connections to the hull and small errors in machining the shoes to equal thicknesses. If the thrust collar is not faced exactly square to the shaft there is a wobbling motion which would result in a pulsating pressure on the oil film for every revolution of the shaft.

The distinctive feature of this design referring to fig. (6) consists in the leveling plates B-1, B-2, whose function is to insure equal division of the load between the shoes.

The leveling plates, B-1, B-2, are all of the same form, and are in effect short levers of equal arms held with slight freedom in the base ring. By reference to the section E-E it is seen that the blocks B-1 fulcrum on their blunt knife edges against the bottom of the channel in the base ring. The blocks B-2, resting upon the blocks B-1, and clear of the bottom of the base ring, in turn support the shoes, contracting through the plugs P. The shoes are free to pivot on their supports as at (Z) to provide for the formation of the lubricating film without reference to any tilting of the plates B required for equalizing. It will be evident from inspection that the parts are in equilibrium only when the shoes receive equal pressure from the collar, except as affected by frictional resistance to the required tipping of the leveling plates B-1, B-2, to compensate for slight errors as for example differences in the thickness of the shoes. Since, however, the contact points as at X, Y, Z, lie in the same straight line if the construction and alignment are perfect the slight tipping of the leveling plates required for overcoming slight errors involve nearly true rolling with every little sliding friction. Therefore the resistance to the automatic adjustment is slight and the division of the load between the several shoes is nearly equal.

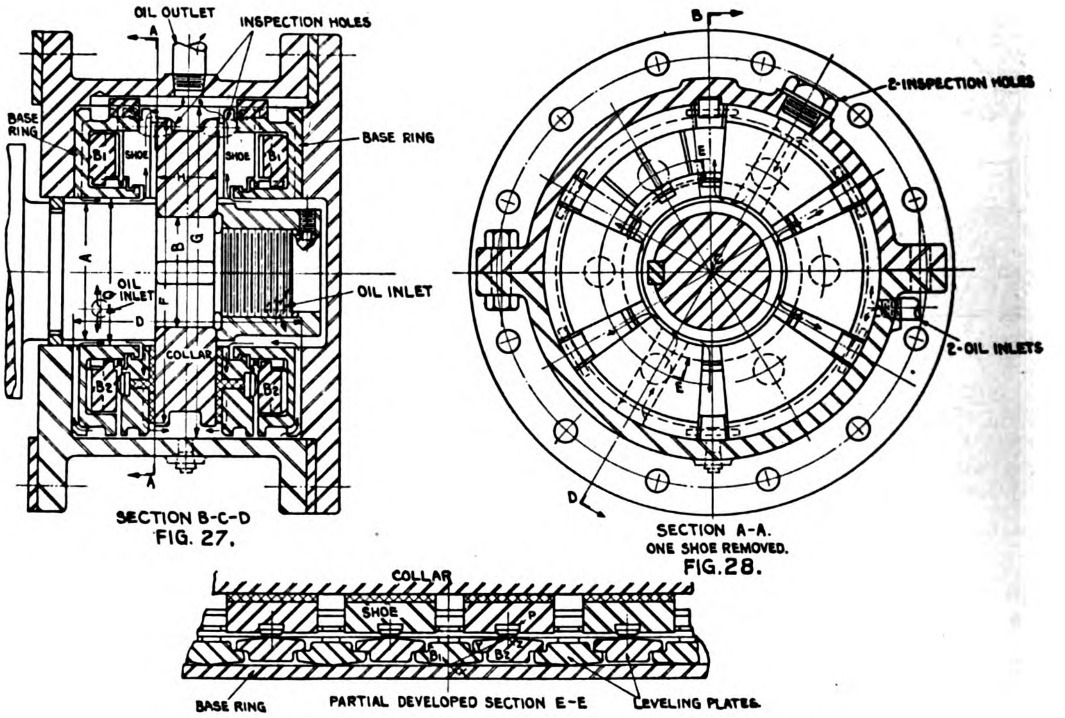

Fig. 6. Kingsbury thrust bearing with equalizing pads

The circulation of the oil in the bearings is shown by arrows.

The extent to which the tilting shoe type of bearing has been used in stationary practice is indicated by the following:

In the plant of the Cedars Rapids Mfg. & Power Co. Montreal, Canada, there are ten vertical hydro-electric turbine generators with Kingsbury thrust bearings. The thrust load on each bearing is 550,000 lbs. and the thrust collars are 61 inches in diameter. Lubricating oil circulates at the rate of 15 gallons per minute with a rise in temperature of about 4° cent. The normal speed is 55.6 rev. per min. but the motor will slow down to 5 r.p.m. before the oil film breaks. Each turbine develops 10,500 h.p. A Kingsbury thrust bearing in a plate glass grinding machine is in constant use loaded to a unit pressure of 980 lbs. per sq. in. and has showed no appreciable wear after five years of service.

When applied to the Diesel engine the single collar thrust bearing greatly shortens the overall length of the engine. Thus the space saved for cargo hold over that allowed by a steam plant becomes still greater. Any expedient which produces an increase in the cargo capacity of a vessel of given displacement and at the same time fulfills its duty better than the old way will undoubtedly be adopted by all builders and owners. It may be mentioned that of the firms manufacturing Diesel engines, the McIntosh & Seymour Corpn. and the Price-Rathbun Engineering Co. have already adopted thee Kingsbury thrust bearing as a standard fitting. The feature of this application is that the engine flywheel is aft of the thrust. Thus a bearing is provided on each side of the thrust collar.

The horizontal marine type which is illustrated in Plate "A" is a Kingsbury installation in the gear casing of a double reduction drive turbine. The figures at hand indicate that over 36 bearings of this type have been fitted for marine purposes. The unit load is from 250 to 380 lbs. per sq. in. total load 32,000 to 89,000 lbs. on single collars from 17 to 27 inches in diameter.

The information and comments on the Michell bearing were taken from the recent paper by J. Hamilton Gibson, Member, read before the Institution of Naval Architects, April, 1919.

Rivalry Among the Crews of Auxiliary Motor-Schooners

Three Motorships to Demonstrate Increased Earning Power Over Sister Ship Carrying Full Sail

CONSIDERABLE interest is being shown by the shipping men of the Pacific Northwest in the three motor-auxiliaries which have lately been completed in that region and placed in commission. They are practically the same size and are similarly equipped. The interest taken is due largely to the fact that these ships will carry, on their maiden voyages, equal cargoes in tended for the same destination. The crews of the ships have made up a purse which will be collected by the vessel making the voyage in the shortest time. A sister ship of identical construction which will not be equipped with auxiliary power is also entered in the race, being given a handicap of 14 days. Two of these vessels, the "Mount Hamilton" and the "Mount Whitney," were built by the McAteer Shipbuilding Company of Seattle, and the remaining two, the "Mount Shasta" and "Mount Hood," were built by the Matthews Shipbuilding Company of Hoquiam, Wash. The latter company has built over sixty large wooden vessels in the last twenty years and is regarded as one of the most successful concerns on the Pacific Coast. All four ships are owned and will be operated by the Cornelius Bull Rederis A/S of Christiania, Norway. The resident owner in Seattle is Mons. Isaacson who has superintended the construction, the installations and the loading of the vessels for their first ventures.

Skandia motors of 240 b.h.p. each drive the twin propellers of the three auxiliary schooners and they are equipped with oil engines for running some of the auxiliaries. The machinery is practically identical on all the ships.

The "Mount Whitney" carries full sail and has no auxiliary power. The following information regarding these vessels will be interesting:—

Fig. 1. M.A. "Mt. Hamilton," with three sister ships one of which will have no auxiliary power. will carry lumber to England from Puget Sound

The "Mount Hamilton' is a four masted tops'l schooner built to Lloyd's specifications and on her trial trip July 14th she averaged a trifle better than 8 knots for over five hours. It was noted that the Skandia Pacific Engine Company, builders of the main engines have discontinued operating the air-compressors off the main engines. The only attached auxiliaries were the circulating and bilge pumps. This permits considerable simplification in design. Air is furnished by two independent compressors. The main compressor, a two-stage 8-inch x 6-inch x 6-inch Rix, made by the Rix Compressed Air & Drill Company of San Francisco, is gear driven by a 16 b.h.p. Skandia engine. The auxiliary compressor, a 6-inch x 6-inch Rix of the single stage type is direct connected to a 13 b.h.p. Skandia engine, both engine and compressor being mounted on a common bedplate. This 13 b.h.p. motor is also geared to a 6-inch x 10-inch McGowan Duplex fire and bilge pump. The auxiliary fire and bilge pump is a 7½-inch x 5-inch x 6-inch Worthington Duplex and may be driven either by steam from the donkey boiler or by air from the main compressor. Both of these pumps are connected so as to deliver circulating water to the main engines. The oil trimming pump is also a Worthington, a 5¼-inch x 4¾-inch x 5-inch Duplex. The electric generating set is a 6 kw., 120 volt General Electric generator direct-connected to a 9 b.h.p. Skandia engine. This also charges the 100-cell Edison storage battery which is installed. A DeLaval oil separator is used for clarifying lubricating oil—the reclaimed oil is used on thrust bearings, pumps, winches, etc.

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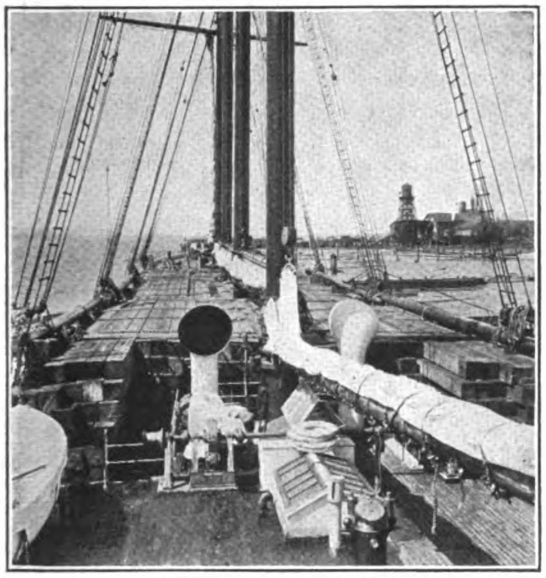

Fig. 2. M.A. "Mount Hood."—Showing deck load of lumber for Great Britain

Other equipment installed aboard the "Mount Hamilton" consists of the following:

A Dake Steam Anchor Windlass.

A 10 h.p. Type "Z" Fairbanks-Morse motor for hoisting sail.

A donkey boiler made by the Eureka Boiler Works of San Francisco.

A Staples & Pfeifer Oil burning system, Marine Pipe & Machine Works representatives for the Northwest.

A feature of the "Mount Hamilton" is that the ship is fitted with exhaust coolers in which the exhaust gases pass through a water spray, thus eliminating sparks and carbon. This also reduces the noise of the exhaust to a minimum. These coolers do away with all danger of sails catching fire from the exhausts. The C. C. Terry Company of Seattle had charge of designing the machinery layout and also the installation of the main and auxiliary equipment. They also superintended the trial trip. This company has within the past few years entered the general field of marine work and are engaged principally in the construction of electrical marine auxiliaries. The "Mount Hamilton" has a fuel capacity of 1,000 barrels of oil, the tanks being supplied by the National Steel & Equipment Company of Seattle.

The following are the dimensions of the "Mount Hamilton" and the "Mount Whitney": Length 250 ft.; Breadth 44 ft.; Moulded depth 21 ft. On their maiden voyages these two vessels will be loaded with 1,500,000 feet of lumber at Bellingham, Wash.

The "Mount Shasta" and the "Mount Hood" are four masted sister ships and have similar power and dimensions to the "Mount Hamilton" and are built to the specifications of Bureau Veritas. The "Mount Shasta" sailed for England July 1st with 1,500,000 feet of lumber. The "Mount Hood" was scheduled to sail about the 20th of July for the same port with a similar cargo. The main and auxiliary motors as well as the other equipment was installed by the builders of the ship. These vessels on their trials each averaged better than 8 knots. The accompanying photograph of the "Mount Hood" gives an idea as to the size of load these vessels will carry and speaks well for the builders in that the ships trim so well. Some of the auxiliary machinery installed aboard these two vessels aside from their main engines is the following:

A 9 b.h.p. Skandia driving a 6 kw General Electric generator. This motor also generates power for the charging of the 100 cell Edison Storage battery outfit.

A 13 b.h.p. Skandia motor direct-connected to a two-stage 8-inch x 6-inch x 6-inch Rix Compressor.

A 13 b.h.p. engine drives a 6-inch x 6-inch stage Rix Compressor and is direct connected to a 4-inch x 6-inch Goulds fire and bilge pump made by the Goulds Manufacturing Company of Seneca Falls, New York. This company also supplied the small hand pump on the after deck which is used for drinking water.

A Worthington pump is also installed to pump oil from the main to the day tanks and for trimming the oil tanks. The Worthington company also supplied an auxiliary pump to be used for fire or for the bilge or for circulating water for the main engines. This pump is driven by air from the compressors or by steam from the donkey boiler.

A two cylinder 25 h.p. Union distillate motor is installed on the after deck and is used for hoisting sail.

It will be remembered that mention was made in our July issue of the competition arranged between these four vessels. At that time we understood that only two of the four vessels were to be equipped with auxiliary power. The "Mount Hood" and "Mount Shasta" are not the boats by those names which have been previously described in "Motorship" and are operated by Gaston, Williams and Wigmore of New York City.

Mention was made of the other "Mount Shasta" changing her name to "Mount Baker" in the August issue.

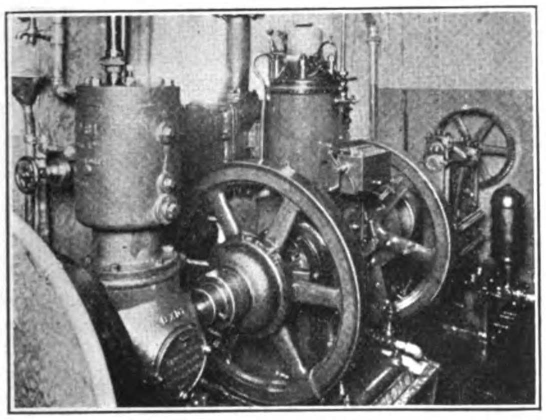

Fig. 3. 13 b.h.p. Skandia engine direct-connected to 6 in. x 6 in. Rix air-compressor and geared to a Gould Fire Pump

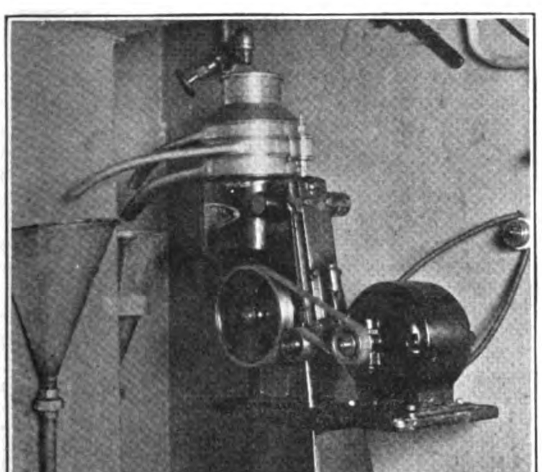

Fig. 4. De Laval Oil Clarifier belt driven by an Emerson shunt motor. By this means the used lubricating oil is made suitable for continued circulation

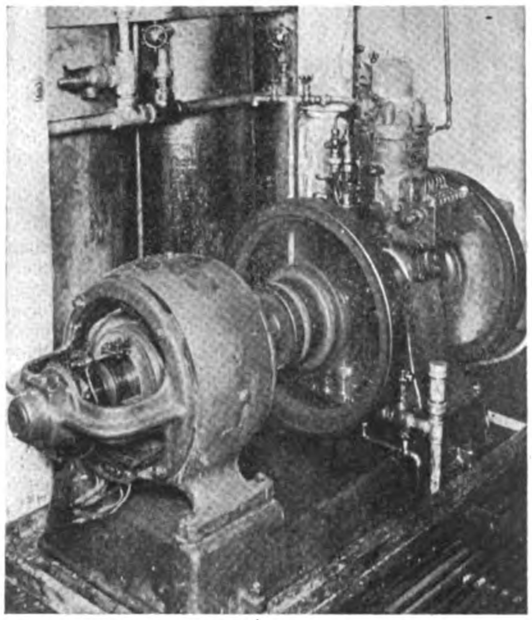

Fig. 5. 9 h.p. Skandia engine direct connected to a 6 K.W. 120 volt G.E. generator for lighting and charging 100 cell Edison storage battery

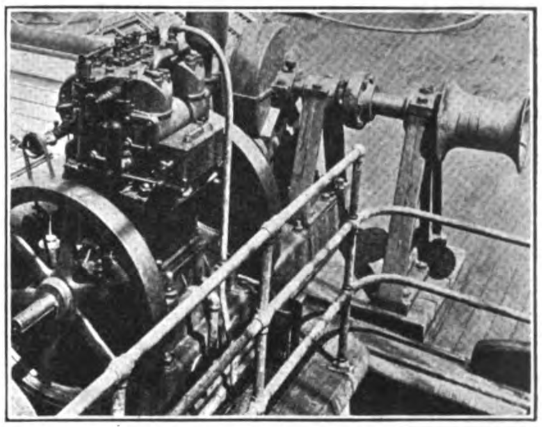

Fig. 6. 2 cyl. 25 h.p. Union engine for handling sails on the "Mt. Hood"

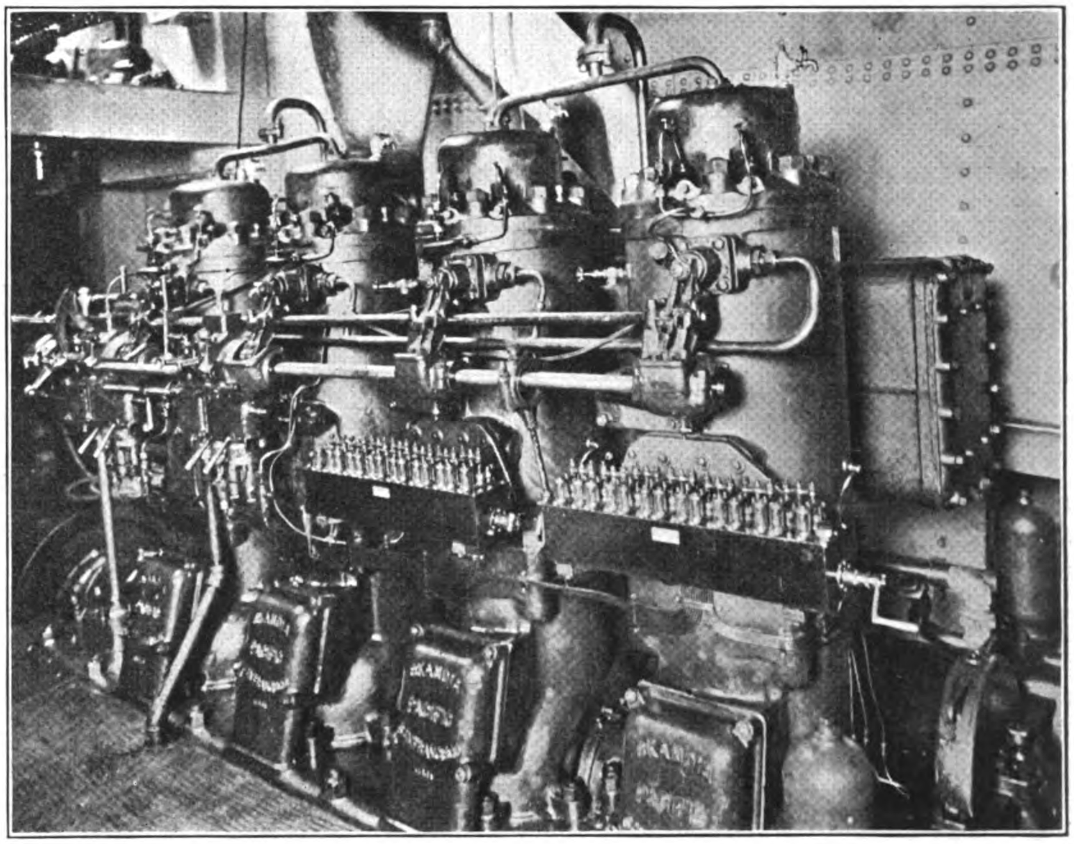

Fig. 7. Port engine of the "Mt. Hamilton"—a 240 b.h.p. Skandia. The "Mt. Hood" and "Mt. Shasta" are similarly powered

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Control of Diesel Engines

Maneuvering Mechanism of Various Well-Known Designs

By DAVID BRUCE

As the operation of a Diesel engine is so different from that of the steam engine it is not surprising that misleading and inaccurate ideas concerning it exist. This is particularly so in regard to such complicated operations as those of starting and reversing. This is, of course, accentuated to a large extent by the multiplicity of designs in vogue. The object of this paper is to give a brief resume of the subject, showing how the systems are developments from various root designs.

When dealing with the starting operation only it is to be remembered that the ignition of the fuel is effected solely by means of the heat generated in the cylinder, due to the rapid compression of the air charge. In order to start a Diesel engine some means must be found whereby the engine shaft can be revolved at a speed high enough to generate sufficient heat to ignite the first charge of oil. Assuming that the engine has been stopped for some time and that all the parts are cold, some of the heat of compression will be dissipated through the walls of the combustion space, so that it is usual to give the shaft two or three revolutions before injecting oil into the combustion space.

The usual way of obtaining these initial revolutions is to admit highly compressed air into those cylinders in which the pistons have just passed the "in" centre. This air supply is cut off before the exhaust period commences. After a few revolutions have been made, the starting air supply is cut out altogether and the fuel pumps and valves thrown into operation. Attempts have been made from time to time to introduce other starting mediums but so far air starting has been retained in every case.

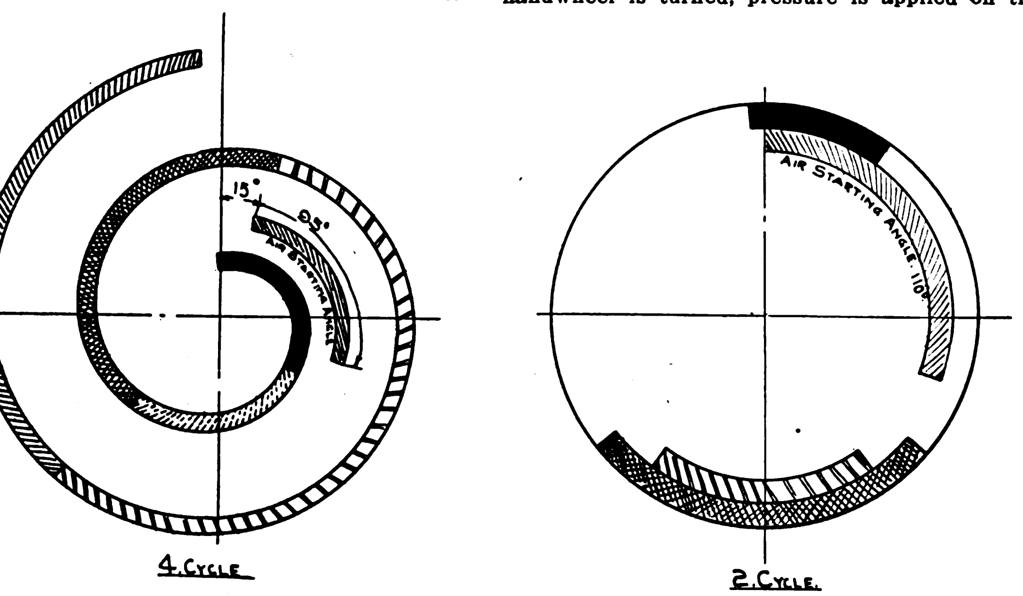

Fig. 1.

Considering air starting arrangements we find there are two distinct ways of using the air; first, by admitting air for a certain period into the working cylinder as mentioned above; and second, by admitting it into some auxiliary cylinder such as the scavenging pumps.

Again there are two entirely distinct methods of controlling the air; first, by mechanically operated valves, etc.; and second, by pneumatic systems.

To illustrate the differences arising out of these divisions the practice of various firms will be dealt with. Before doing this, however, the principles underlying the operations are given below.

To be efficient a starting system should be capable of starting the engine with the crank shaft in any position, without the necessity of having to bar the shaft around. In engines of only 1, 2 or 3 cylinders this is impossible, but in engines of four or more cylinders the starting angle can be made large enough to insure at least one cylinder being in a starting position.

As compressed air is expensive to produce the starting system must be carefully designed. The commencement of the starting period is usually a degree or two after the crank has passed the inner centre, and it continues until a few degrees before the commencement of the exhaust period. A glance at Fig. 1 will explain this as applied to both two-cycle and four-cycle engines. It will be seen that in both cases the starting period extends over an angle of 90° thus insuring the starting of a four cylinder engine from any position.

To reduce the amount of work to be done in starting it is usual to release the compression in the cylinders before commencing the operation so that there will be no unnecessary resistance. Another important point is that there must be no fuel charges injected during the first two or three revolutions or there will be an accumulation of oil on the walls of the combustion space. Oil injected without immediately burning will tend to carbonize, or to give rise to a dangerously high explosion pressure, at the first ignition. The fuel is not injected until the engine is ready for it, i.e., after two or three revolutions and if the engine does not pick up immediately, the fuel injection gear must be thrown out before attempting to restart. In most systems this is provided for in the design of the starting gear so that the system is fool proof in this respect.

In multi-cylinder engines it is usual to throw half the cylinders on fuel at first leaving the remainder running on air until the first set commences firing, then the remainder are thrown onto fuel.

To relieve the compression in the combustion space various methods are employed, the cheapest and simplest being that in which the indicator cocks (one in each cylinder) are opening before starting, and closed immediately upon turning on the starting air. In pneumatic systems it is customary to have a small piston on the cylinder relief or exhaust so that as soon as the starting handwheel is turned, pressure is applied on this piston opening the valve and releasing the compression. A further movement of the handwheel allows these valves to close.

Turning our attention now to the actual systems, we find that one of the more general type is as shown in Fig. 2 worked in conjunction with the valve shown in Fig. 3. Referring to Fig. 2 it will be seen that mounted on the cylinder head are two pillars supporting a stationary shaft. Mounted loosely on this shaft are the levers (A) and (D) operating the suction and exhaust valves respectively. The hand lever (E) has an eccentric sleeve attached on which are mounted loosely the levers (B) and (C) which operate the fuel injection and starting valves. This hand lever is provided with a spring catch engaging with the fixed washer (F). The eccentricity in the levers (B) and (C) is arranged so that these two valves cannot be in operation at the same time. Fig. 2 shows the disposition of these levers with the hand lever in the starting position. The fuel injection lever (B) is clear of the cam, while the starting lever (C) is in contact with its cam, thus the cylinder will receive starting air but no fuel. The running position is shown in Fig. 2b with the starting lever clear of the cam, and the fuel injection lever in contact. Referring to Fig. 3 it will be seen that the valve is of the inward opening type, mounted in a cage (B) and held on its seat by a spring (E) the starting lever operating between the collar (D). Air is admitted to the annular space (C) from the air reservoir and when the valve

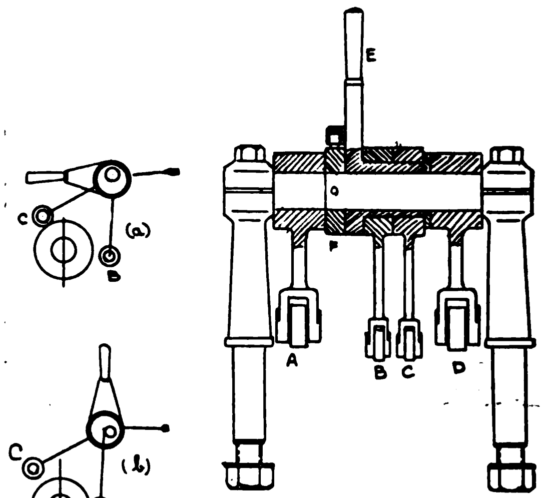

Fig. 2.

is opened the air flows through into the cylinder.

This system which provides for individual control of any cylinder is universally used for land installations but for marine purposes a more centralized system is desirable. One such system which is both mechanical and pneumatic is shown in Figs. 4, 5, and 6. Referring to Fig. 4 which illustrates the arrangement for a twelve cylinder engine, it will be seen that two master valves are provided, each controlling the supply of air to a starting box. Each of these boxes distributes the air at the correct time to starting valves in the cylinder heads.

To start the engine both master valves are opened allowing starting air to pass into the chambers (A) Fig. 5 in the starting box. In this box are a number of poppet valves, the stems of which are normally held off the cams by light springs. When the starting air is turned on the pressure above the valves suffices to hold them on to the cam. As the starting angle in this engine is 95° it follows that at least three of these valves will be held off their seats by the cams and starting air will pass these valves to their respective valves in the cylinder head Fig. 6. These valves are automatic, the valve (A) being of the usual inward-opening type held on its seat by the spring (B). The air entering the valve forces it open and enters the cylinder until such time as the poppet valve cam in the starting box allows the valve (B) Fig. 5 to close. When the engine has made the usual number of revolutions one master valve is closed and the fuel injection gear for the corresponding cylinders is put into operation. This operation is repeated later for the remaining cylinders.

As an example of a pneumatic system and also of one using the scavenging pump as a starting cylinder the M.A.N. system will be considered. (See Fig. 7.) To start the engine the valve on top of the reservoir is opened admitting air to the valve (A) and by way of a small passage (B) to the underside of the valve (A) and so to the upper side of the valves (D). The hand lever (C) now being moved into the starting position, the valve (D) is opened allowing the pressure below (A) to escape to atmosphere which by reason of the excess pressure above it now opens shutting off the passage (B) and allowing starting air to pass to the space (F) in the cylinder head, and to the space above the valve (E). This valve (E) being open due to the position of the hand lever allows the air to pass to the slides (G) forcing the side out until its roller is in contact with its cam.

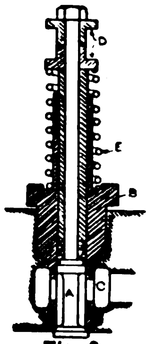

Fig. 3.

Assuming this cam to be in the starting position the air will pass by way of the port (H) to the piston (J) in the cylinder valve. The pressure on this piston forces the valve open and allows the air in the space (E) to enter the scavenging cylinder until such time as the cam moves the slides (G) into a posi-

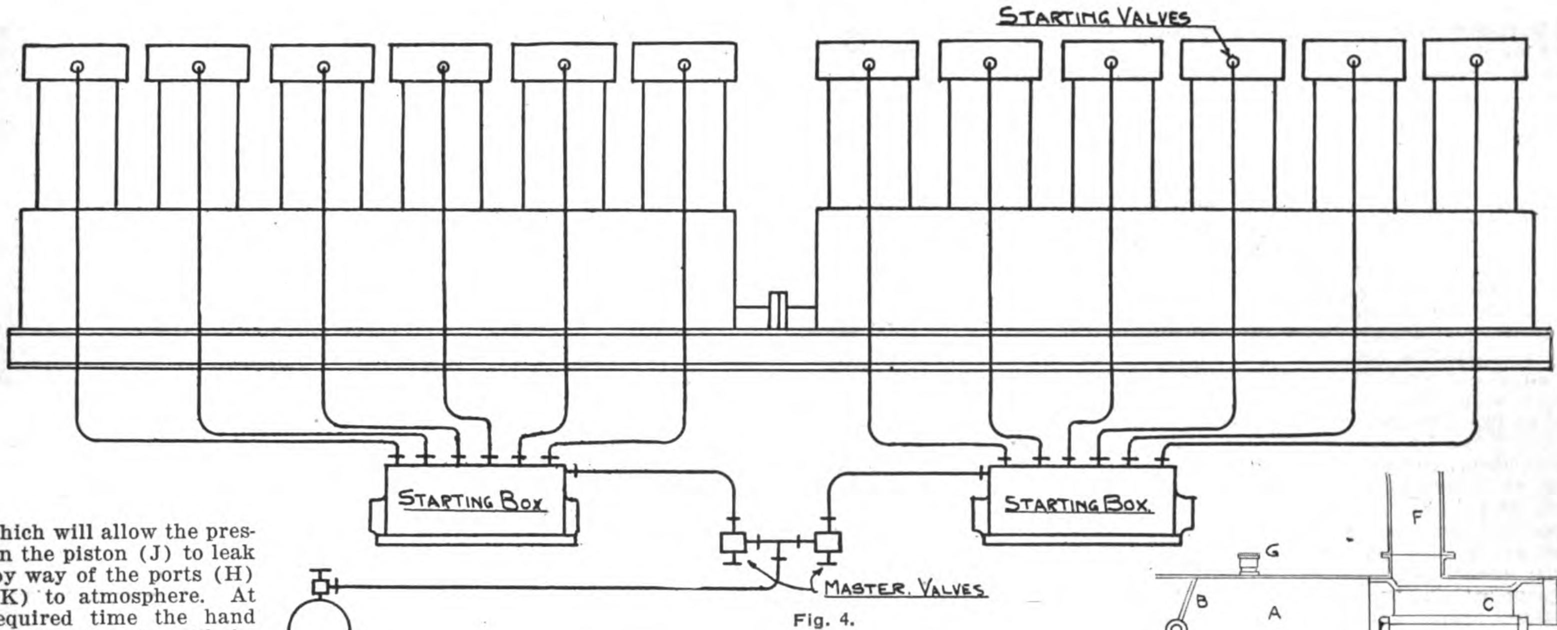

Fig. 4.

tion which will allow the pressure on the piston (J) to leak back by way of the ports (H) and (K) to atmosphere. At the required time the hand lever is moved back until the valves (D) and (E) are both closed and the hand lever is in the fuel position, the fuel pumps are put into operation by the rod (L). In the Polar engine the designers have also

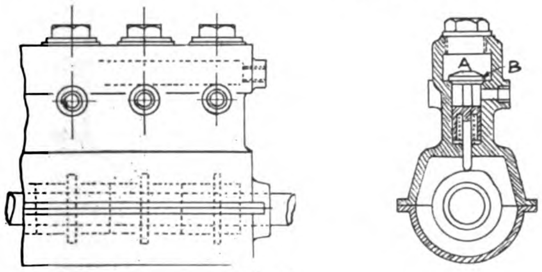

Fig. 5.

made use of the scavenge pumps as starting cylinders. In this case the scavenge pump suction and delivery are controlled by mechanically operated piston valves, and these valves are also used in starting the engine. As arrangement of these valves is given in Fig. 8 in which (A) is the scavenge air suction main, (B) is butterfly throttle for blanking off the main (C) the master valve for reversing purposes, (D) and (E) the piston valves governing the suction and delivery from the pump, and (F) the connection to the scavenging ports. Consider first the working of these valves with the engine running normally on oil, and with the master valve (C) set for ahead running as shown at (A). The valve chest shown is used to control two adjacent scavenge pumps having their cranks set at 180° and with the valves in the position (a) but with the throttle

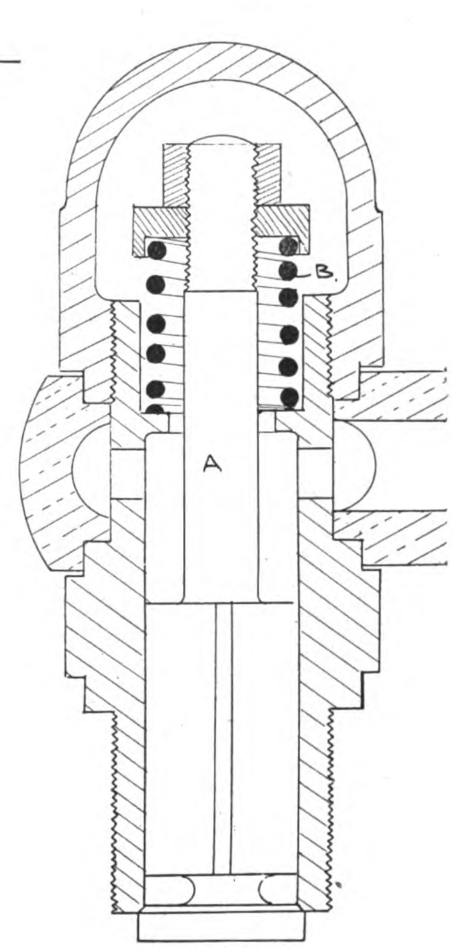

Fig. 6.

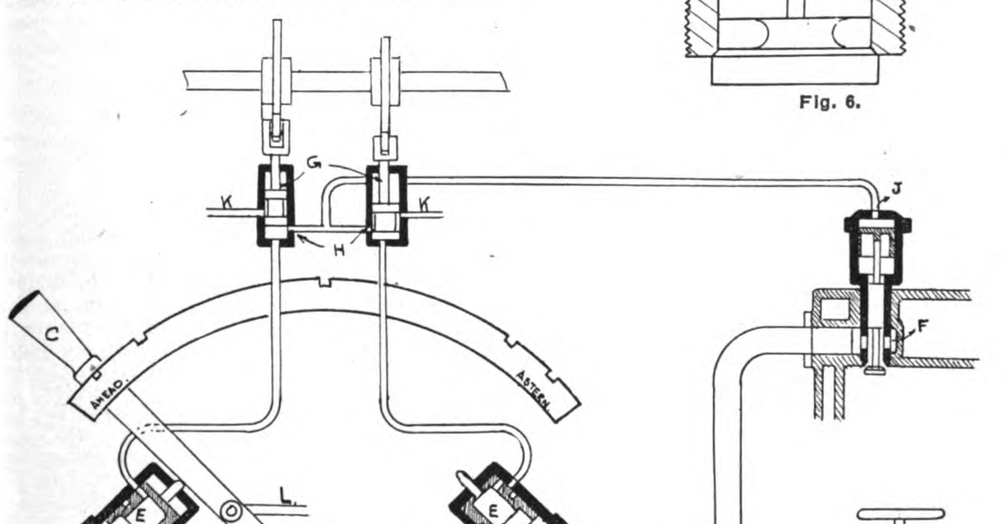

Fig. 7.

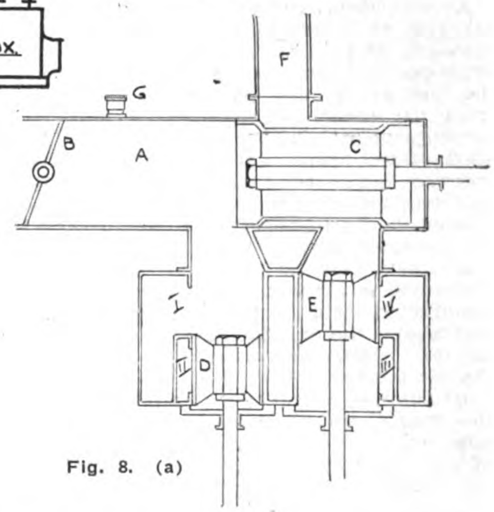

Fig. 8. (a)

(B) open, one scavenge pump will be drawing air from the suction main by way of the ports (1) for one whole stroke, while the second pump will be delivering by way of the ports to the supply pipe (F). During the next stroke when the valve (D) has moved up and the valve (E) has moved down, the second pump will draw its supply from the main (A) by way of the ports II while the first pump will be delivering its supply by way of the ports IV to the supply pipe (F).

For starting purposes the operation is as follows: By means of a hand lever the throttle valve (B) is closed and the starting valve (G) is opened allowing high pressure air to flood the valve chamber. Now either the ports I or II will be open and the starting air will pass to one of the scavenge pump pistons, and force the piston outward. At the end of the outward stroke a small port in the scavenge pump cylinder is uncovered allowing the starting pressure in the scavenging cylinder to fall to that of the atmosphere. On the return stroke the air above the scavenging pump piston is delivered by way of the valve E to the scavenge pipe F. In this design the oil can be admitted without cutting out the air supply and more reliable starting is insured. As soon as the engine has picked up on oil, the throttle valve (B) is opened and starting valve (G) is closed.

The method of using the scavenge pump as a starting cylinder is beneficial in that the starting air, which becomes extremely cold upon expanding is not injected into the working cylinder, the walls of which should be kept warm. This is more important when reversing is attempted for when the cylinder walls are extremely hot the projection of extremely cold air gives rise to dangerously high temperature stresses.

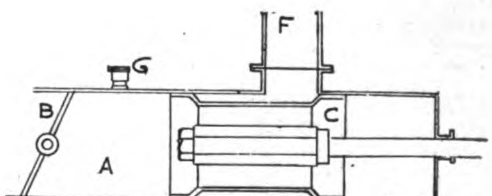

Fig. 8. (b)

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Starting Air Quantity and Pressure.

With regard to the quantity and pressure of air required for starting purposes the following figures may be of interest.

A maneuvering test was made of an engine of 750 B.H.P. using a pneumatic system similar to that shown in Fig. 7. This engine had a stepped piston for supplying the scavenge air, and this piston was also used for starting purposes. For an installation having 2-750 b.h.p. engines the total air storage was 42.5 cubic feet in five reservoirs. Four of these reservoirs had air at 1,000 lbs. per sq. in. while the remaining one was supplied from these four through a reducing valve and contained air at 625 lbs. per sq. in. Under these conditions thirty successful starts were made. With 800 lbs. pressure in the main tanks seventeen starts were made, the pressure in the reservoirs after the seventeenth start being 144 lbs. per sq. in. This was not sufficient for another start.

A 400 b.h.p. engine having a total storage capacity of 21 cubic feet in two reservoirs, at a pressure of 650 lbs. per sq. in. was also tested. With one of these reservoirs containing air at 295 lbs. per sq. in. and having its supply re-newed from the other reservoir which had an initial pressure of 650 lbs. per sq. in., twenty successful starts were made. During these operations the pressure fell to 103 lbs. per sq. in., which was insufficient for another start. This engine could be started from "all cold" in 10 seconds, or it could be re-started from "warm" in 5 seconds.

A comparison of the time required to get a motorship under way from all cold with the time required to get a steam vessel under way from all cold shows the great advantage possessed by the former vessel. Assuming that in the motorship the air storage bottles are empty and that these must be re-charged by the auxiliary compressor, the vessel can be under way in less than ten minutes. [This feature depends on the capacity of the auxiliary compressors. Ed.]

Reversing Systems.

The theoretical conditions to be overcome in reversing a Diesel engine present a more difficult problem than that of starting, but as far as the practical construction goes the mechanism is quite as simple.

When the engine is running in the ahead direction a definite sequence of events is taking place in the cylinders as shown in Fig. 1, both in a four-cycle and a two-cycle engine. To reverse the engine, it is necessary to reverse this sequence in relation to the crankshaft. This sequence is controlled by the cams on the camshaft so some means of adjusting the camshaft must be adopted. For illustration let us consider fuel injection period shown in Fig. 9. This commences five degrees before the inner center and continues for thirty-five degrees after the center. The center point of this period will therefore occur fifteen degrees after the center. To reverse this period it is necessary that the closing point be thirty-five degrees after the center when running in the opposite direction. Therefore, the mid point of the period must be moved to fifteen degrees on the other side of dead center, *i. e.*, a total backward movement of thirty degrees.

In the M.A.N. system this reversing angle is made the same for both the fuel and scavenge valves, that is 30°. As these valves are symmetrical in shape, i.e., the opening and closing profiles are the same, it follows that if the camshaft be turned back through 30° relative to the crankshaft, the engine will be in the correct setting for astern running. The exhaust in this engine being through ports, no account need be taken of exhaust valves.

To obtain the necessary adjustment this firm has adopted an ingenious device. The vertical shaft between the crankshaft and camshaft is made in two pieces connected by a jaw coupling. Between the back faces of the teeth in this coupling a large clearance is allowed. This clearance is equivalent to a backward movement of thirty degrees on the camshaft. There remains only the air starting cam to be dealt with, and as this period is considerably larger than is required for the fuel and scavenge valves, the thirty degree adjustment on the camshaft is insufficient and two cams are necessary; one for ahead running and one for astern running. Referring to Fig. 7 the two cams are indicated and it will be seen that to start the engine in the reverse direction it is necessary only to move the starting hand lever over to the astern position. The operation for starting can then be continued as for running ahead.

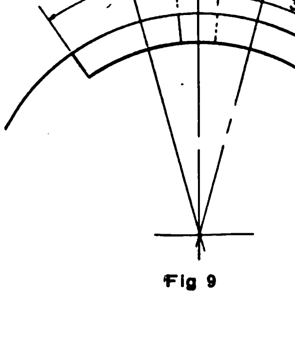

Fig 9

Another and more usual method of obtaining the adjustment of the camshaft is illustrated in Fig. 10. In this case the camshaft is driven from the crankshaft by a vertical shaft, through gear wheels. Owing to the spiral angle of the teeth, if the vertical shaft be moved axially, a definite angular displacement must take place between the camshaft and crankshaft. This displacement is easily calculable for a definite axial travel of the vertical shaft. This travel can be obtained either

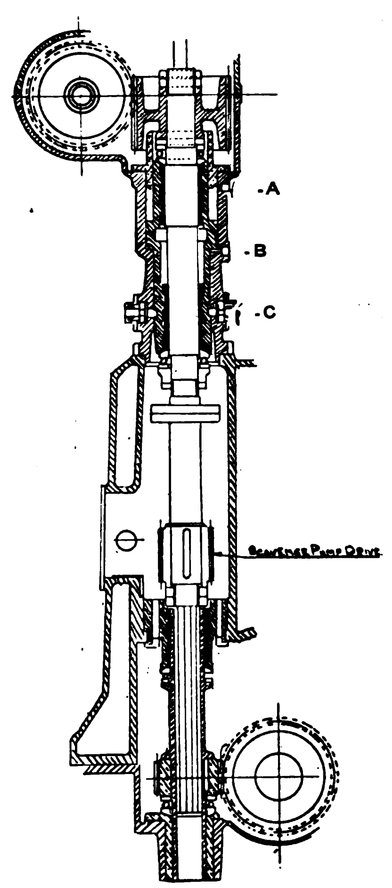

Fig. 10

by mechanical or pneumatic control. Fig. 10 illustrates the Fiat method of employing an air cylinder for this purpose. The inlet and outlet for the maneuvering air may be at A and B. In this design the lower end of the vertical shaft is carried on a hollow sleeve, mounted loosely on the shaft. A key makes the shaft and sleeve rotate together but allows the shaft to slide through the sleeve. At the upper end, the shaft revolves in bearings in the air piston; roller bearings are provided to take the end thrusts of the shaft. The shaft is raised or lowered by admitting pressure either to the upper or lower sides of the air piston as desired forcing it to move into the ahead or astern positions. Catches are provided to lock the shaft in either position. These catches (shown at C) are automatic in their locking action and are provided with a small piston under which pressure is admitted to release the catch before moving the shaft. The piston valves for the scavenging air pump are operated by a special gear on the vertical shaft. The angle of the gear teeth is such that the movement of the vertical shaft insures the correct angular displacement for the scavenge valves.

The Korting Diesel engine provides an example of a mechanical gear for turning the camshafts with respect to the crankshaft. It is illustrated in Fig. 11. The vertical shaft (A) drives two horizontal shafts (B) and (C) through spiral gears. These shafts in turn operate the shafts (D) and (E) respectively. The shaft (D) operates the scavenging pump piston valves, while the shaft (E) is the engine camshaft. As the handwheel (F) is turned the shaft (D) is moved axially by means of the screw (G) and lever (H) so that the camshaft is displaced angularly in relation to the crankshaft. Meanwhile the shaft (B) is also being moved axially by means of the screw (G1) and the lever (H1) effecting the angular displacement of the Shaft (D).

The Polar Diesel engine has a different reversing arrangement. Referring back to Fig. 8-b, the position of the valves is given for astern running. It will be seen that the master valve (C)

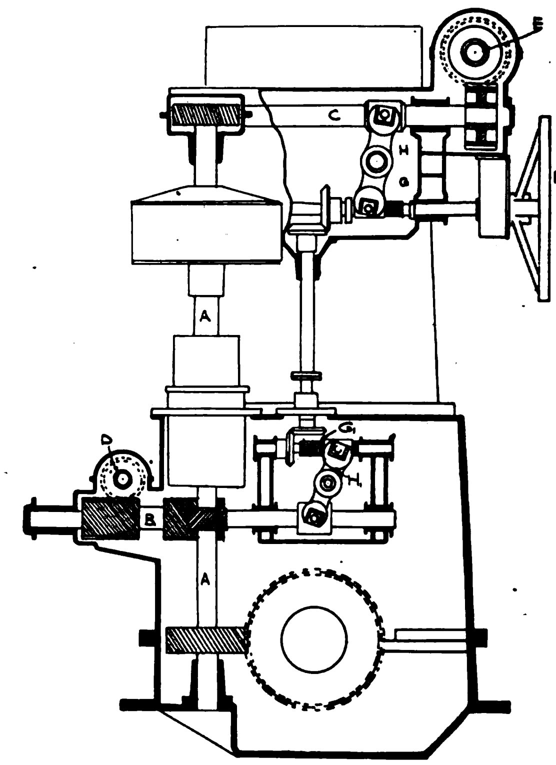

Fig. 11

has been moved into the left hand position, throwing the suction main (A) into communication with the ports III and IV and the delivery main (F) into communication with the ports I and II. Thus the suction stroke of the scavenge pump for ahead running is now the delivery stroke, and vice versa. Thus when the starting operation is performed for astern running the engine will of course rotate in the astern direction.

The more difficult problem is in a four-cycle engine. In such engines cams are required for the suction, exhaust, fuel injection and air starting valves, and as each of these valves has a different reversing angle, it is obvious that the problem cannot be solved by a definite angular displacement of the camshaft. Following out this assumption, we come to the conclusion that there must be two sets of cams, one set arranged for ahead running and the other arranged for astern running. Therefore change-over gear must be provided so that the valves can be operated by either set as desired.

For instance, the Werkspoor engine illustrated in Fig. 12 has two entirely different camshafts, one having cams set for ahead, and the other for astern running. These two shafts are mounted in a series of brackets pivoted on a main shaft and provided with balance weights. This main shaft is driven from a lay shaft by two long eccentric rods, the lay shaft being introduced to obtain the 2 to 1 ratio. A worm and quadrant is provided for swinging the camshaft over. This gear, in the form described, or slightly modified, has been fitted to all Werkspoor direct reversing engines until recently. On account of the decreased weight and lower cost a new design of gear has been adopted. This new gear was described and illustrated in the "Motorship" Patent Record for February, 1919.

The Burmeister and Wain reversing gear is another excellent solution of this problem in four-cycle engines. It is illustrated in Fig. 13. The camshaft is driven from the crankshaft by a train of spur wheels geared to give the 2 to 1 ratio. The camshaft in this design is not placed at the cylinder head, but is kept low in order to reduce the gearing required. As a result long push rods are utilized. The lower end of these push rods are linked up to a lay shaft, which is used for reversing purposes only and does not rotate when the engine is working. Mounted on the shaft carrying one of the gear wheels is a drum provided with a deep groove having an offset as shown in the expanded view (13a). On this shaft is also an eccentric and rod, the small end of which is connected to the lay shaft crank. This

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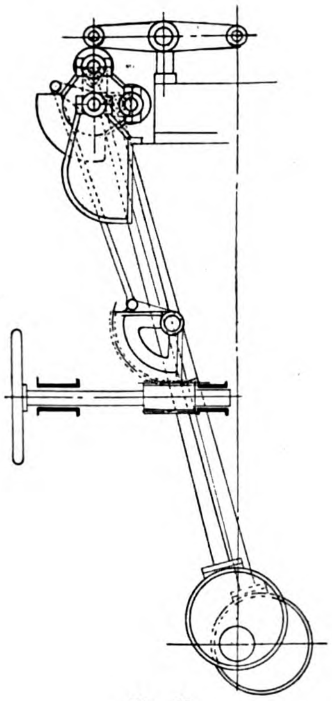


Fig. 12

reversing shaft is also stationary, while the engine is running. Mounted between the camshaft and reversing drum, is a dog lever, the end engaging with the drum being guided in the slot, while the end engaging with the camshaft is held between two collars. The camshaft is provided with two sets of cams. The operation of reversing is as follows:

When the drum shaft is revolved, the eccentric moving outward cases the lay shaft to turn enough to swing the push rod ends clear of the cams. Meanwhile, the reversing drum has also turned so that the end of the dog lever is now at the beginning of the offset in the groove. The shaft continuing to turn, the end of the dog lever is forced to move laterally to the obliquity of the groove. This lateral movement is transmitted to the camshaft and continues till the astern cams are in position, i.e., when the final point of the oblique path is reached, the remainder of the operation consists in letting the eccentric push the rods back on the cams.

There are several other methods of moving the shaft, but this example will suffice to illustrate the principle.

The Vickers system is illustrated in Fig. 14. The camshaft (A) is provided with two cams for each valve, one for ahead and astern running. Below the camshaft is a lay shaft (C) having an eccentric sleeve below each cam on which is mounted a lever (B). The valves are operated by a long push rod (D) one for each valve. The lower ends of these rods are guided. The two levers for any valve operate the same guide. The eccentrics on the lay shaft are set at 180° so that one lever for each valve is in contact with its cam, while the other is clear of the cam. To reverse the engine, all that is necessary to do is to turn the lay shaft through 180° thus bringing the astern cams in operation.

It is necessary that reversal should be a practically instantaneous operation if it is to be of practical value in an emergency. Every effort must be made by the designer to reduce the amount of mechanical work to a minimum, and to substitute pneumatic gear, or something similar, in place of hand gear. This is usually done by fitting an air or oil motor such as described in the Fiat gear shown in Fig. 10.

With an efficient design the time taken for full reversal should not be more than 10 to 15 seconds. Tests of an M.A.N. engine of 950 B.H.P. gave the following results: Full reversal from 260 revs. per min. ahead to 240 revs. per min. astern in 10 seconds. When four reservoirs were used having a total capacity of 41.5 cubic feet of air at 500 lbs. per sq. in. the engine was reversed 14 times, the pressure meanwhile falling to 85 lbs. per sq. in. which was insufficient for another reversal.

It will be seen from these figures that the reversal of a Diesel engine is not as uncertain or slow as is generally supposed.

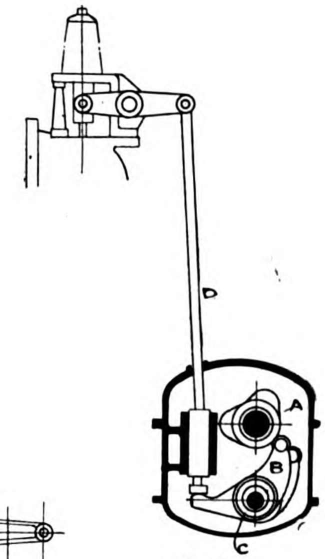
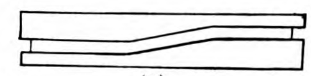


Fig. 14



(a)

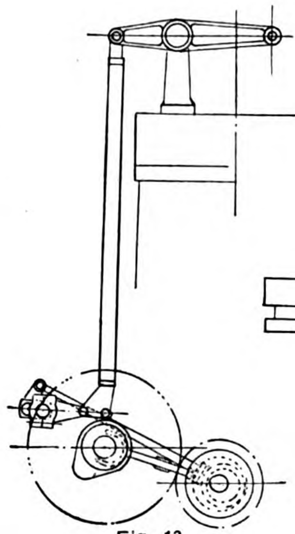
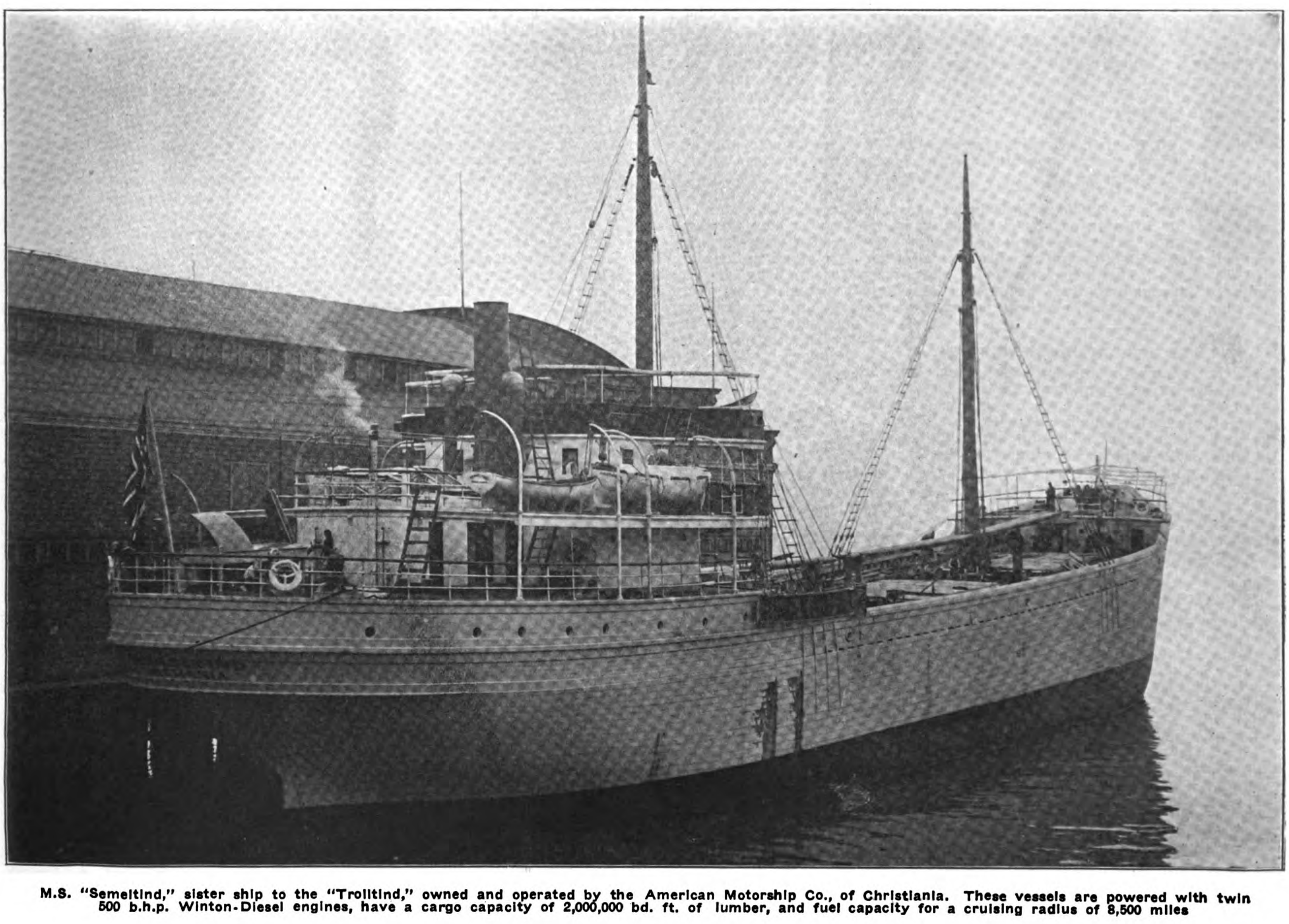


Fig. 13



M.S. "Semeltind," sister ship to the "Trolltind," owned and operated by the American Motorship Co., of Christiania. These vessels are powered with twin 500 b.h.p. Winton-Diesel engines, have a cargo capacity of 2,000,000 bd. ft. of lumber, and fuel capacity for a cruising radius of 8,500 miles

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Interesting News and Notes from Everywhere

FRENCH FUEL-OIL IMPORT DUTIES REDUCED

A bill was adopted on July 22nd by the French Senate reducing the import duties on fuel-oil to a level of those on coal. Up to that time the duty on oil for Diesel engines had been nearly 80 times that of the duty on coal. The new ruling should greatly encourage the use of the oil-engine in France, particularly as coal is very scarce and expensive.

THIRY-TWO BRITISH SUBMARINES BUILDING

The British Admiralty has decided to complete 32 of the Diesel driven submarines now under construction. Many of these are of high power.

AMERICAN BANKERS PURCHASE INTEREST IN BRITISH-DUTCH MOTORSHIP OWNING COMPANY

Kuhn, Loeb & Co. the well-known New York bankers have purchased 150,000 shares (value about $25,000,000) of the Shell Transport & Trading Co., which company controls the Anglo-Saxon Petroleum Company, owners of the large fleet of motorships illustrated and described in our May 1919 issue. The Shell Co. is the British branch of the Royal Dutch Petroleum Co.

EIGHT NEW FRENCH MOTORSHIPS

Eight engines of 1,500 b.h.p. are under construction at the Harfleur works of Messrs. Schneider & Co. the French engineers. Unless these engines are purchased by other interests they will probably be installed in eight single-screw steel freighters of 3,500 tons d.w.c., now building at Messrs. Schneiders' Harfleur shipyard. These engines are about 30% complete.

SARDINE MOTOR-FISHING-BOATS

During the month of June, the motor fishing-boats of Le Havre, France, only earned 30,000 francs. Bad weather kept them in port most of that period.

A MOTORSHIP FLEET WITH DIESEL-ELECTRIC DRIVE

The Central American Transportation Corpn. is being organized in Chicago for the purpose of operating a line of Diesel electric driven vessels between the United States and the Central American republics.

The General Electric Co. of Schenectady, N. Y. will supply the electrical apparatus and the Jacobson Engine Company of Albany, N. Y. will install the engines.

Mr. Frederick T. O. Wood of Chicago has been appointed Chief Engineer of the company and it is through his extensive investigations of the economics of electric drive that these installations will be made. The vessels will be electrically equipped throughout for propulsion, heating, lighting, cooking, and all auxiliary power purposes.

TECHNICAL ADVERTISING

Mr. R. E. Lovekin, formerly Managing Director of The American Screw Propeller Co. and an authority on Engineering Advertising, has disposed of his interest in the American Screw Propeller Co. and has opened offices at 610-11-12 Penfield Building, Philadelphia, Pa., under the name of R. E. Lovekin, Advertising Engineer.

Mr. Lovekin's experience in both the Engineering and Advertising fields, as well as his wide acquaintance among the shipyards' officials and marine engineers of the country, establishes him in a fortunate position for his new work.

MOTORSHIP ACTIVITY IN EUROPE

A new motorship has been built and is about ready for delivery by the Aktiebologet Gotaverken, Gothenburg, Sweden. This new vessel, M.S. "Balboa," is 425 ft. in length and about 9,500 tons d.w. and is the third of four sister ships to be built for the Rederiaktiebolaget Nordstjernan (Johnson Line). They are all powered with two Burmeister & Wain engines of 2,000 i.h.p., giving a service speed of 12 knots.

The M.S. "Asia" has been launched at Copenhagen by Burmeister & Wain for the East Asiatic Co. and is similar in size and power to the "Balboa" mentioned above.

In Italy, the motorship building has taken on a permanent aspect as over 50 standard full-powered vessels are in course of construction. The vessels are from 6,000 to 11,000 tons deadweight capacity. The success of the two-cycle engine for marine propulsion will no doubt be determined by the service obtained from some of these vessels as about 40 standard 9,000 tons d.w.c. will be equipped with two Fiat two-cycle, four-cylinder motors of 1,150 b.h.p. each. These vessels, which will be owned by the Societa Nagionale di Navigazione of Genoa have an estimated speed of 11 knots and a fuel consumption of 11 tons per 24 hour day.

HEAVY FORGINGS AVAILABLE FOR WEST COAST BUILDERS

The organization of the Pacific Construction and Engineering Company, of Seattle, which took place in 1917, marked the beginning of a new era in northwestern industrial affairs. Up to that time ship builders and other manufacturers of the northern Pacific coast had been dependent upon other parts of the country for their heavy forge work. With the opening of the new plant, the largest heavy forge shop west of Pittsburgh, a complete equipment of furnaces, lathes, presses and handling machinery was placed at their disposal.

Located on the east wateraw, in the heart of the manufacturing district, the plant has both tidewater transportation and rail service. The latter is available through two spur tracks. The main building is 700 ft. long by 120 ft. wide. Its height, 100 ft., gives plenty of room for operating the heavy electric Shaw cranes, six of which are now in service. Other equipment includes three quick-acting steam hydraulic presses, a 2500 pound hammer for the smaller jobs, eight hogging lathes capable of handling the largest forgings, shapers, drill presses, air compressors used for chipping or centering the shafts, and a complete equipment for the testing laboratories. Among the latter is included a large testing machine made by Tinius Olsen, of Philadelphia. A battery of twenty-one heating furnaces flanks the main bay, and is supplemented by two large annealing furnaces.

THE COAL SHORTAGE IN ENGLAND

As a result of the activities of the Diesel Engine Users Association in Great Britain the Coal Mines Department of the Board of Trade will consider applications from communities or factories that use electricity generated by oil engines for the privilege of being exempted from the restrictions of the Household Fuel and Lighting Order. It was urged that it would be contrary to the national interests to place any unncessary check on the output of electrical energy for use in the increased production required in the various industries. It has been acknowledged that the threatened coal famine in England might be relieved by the increased adoption of the oil engine for electrical power plant installation.

The Association also urged that encouragement be given to the method of economical consumption of coal by distillation. This would provide a domestic supply of liquid fuel and also valuable by-products besides coke, which is a clean burning economical solid fuel.

Distillation of coal was Germany's principal source of fuel oil during the war.

BUILDING OIL ENGINES IN CHINA

Writing from Canton, China, under date of March 6, 1919, the American Consul gives some interesting information concerning the manufacture of crude oil engines in that district. His communication is as follows:

"The oil-engine business has been making rapid progress and promises to become an important factor in the industries of Kwangtung. During the recent years numerous flat-bottom launches and native craft, specially built for operation in the shallow rivers of Kwangsi Province and in the Canton Delta, have been constructed and fitted with oil engines which have given satisfactory results.

"From a reliable source it is learned that the firm of Hip Tung Wo, capitalized at $120,000 local currency (about $90,000 United States currency), employs more than 100 workmen, under the supervision of native engineers, in the construction of these oil engines. Certain materials for the manufacture of shafts, such as huge lathes, etc., which can not be made locally, are purchased either in the United States, as evidenced by orders placed with foreign commission houses in Canton or in Japan. In consequence of this the natives are at present able to manufacture shafts only 4 or 5 in. in diameter. Larger shafts, if needed, are made in the Hongkong & Whampoa Dock, Hongkong, or ordered in America.

Prices of Shafts and Engines—Advantages Over Steam Engines

"Prices per 100 lb. for shafts made in the United States are roughly as follows: Diameter of 4½ in., $8.67; 6 in., $8.80; 7 in., $9.25; and 7½ in., $9.35.

The price of a set of oil engines, exclusive of the shaft, locally made and capable of developing 160 h.p., is $22,400 Canton currency (about $16,800 United States currency). Those purchased in Japan range from 30,000 to 120,000 yen ($14,940 to $59,760) for 180 to 360 h.p.

In addition to the use of oil engines for launches, several silk filatures, small weaving factories and rice mills have installed these machines, and they have been working quite satisfactorily. The consumption of fuel oil is ½ lb. per hour for each horsepower, and the price is $35 to $45 local currency per ton. The Asiatic Petroleum Co., which has branches in the interior, is the only firm in Canton supplying this oil.

"The growing appreciation of oil engines in local shipbuilding is undoubtedly due to their increased accommodations for passengers and cargo, as vessels fitted with steam engines require provision for greater space for boilers and coal bunkers and the employment of more engineers. With the addition of the Kwong Nam factory, where more than 1000 workmen are already employed in its dockyard, in which ships up to 3000 tons can be built, there is every reason to believe that the future prospects of the development of oil and steam engines for the shipbuilding industry of Canton are most encouraging."

A typical Danish standardized motor coaster. Hundreds of these economical vessels have been built and powered with about 300 to 500 h.p. motors of either Diesel or surface ignition type

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As Others See Us

Burmeister and Wain Director Expresses His Views on the Motorship in the American Merchant Marine

HOME in his office after his return from the United States the technical head of Messrs. Burmeister and Wain, Director Blache, granted our Danish correspondent an interview regarding current motorship and Diesel engine matters.

Mr. Blache found that his trip to the United States was illtimed as both the Shipping Board and private owners were not at liberty to decide on their new building program for the following reasons:

Congress had not yet granted the additional appropriations necessary to allow the Shipping Board to proceed further with their very commendable decision of adopting Diesel engined motorships as part of the program established by the Emergency Fleet Corpn. Nor had the private owners had their complete fleets, most of which had been requisitioned, returned to them by the Shipping Board and they were still more or less under the control of the U. S. Government.

However, the chartering board of the U. S. Shipping Board had obtained the control of a number of motorships and Director Blache found that the American shipowners were fully aware of the advantages of the motorship over the steamship. This realization of the economic possibilities of the Diesel engined cargoship resulted in a careful and minute study of the arrangements and details of the proposed vessels when the scale drawings were published by "Motorship."

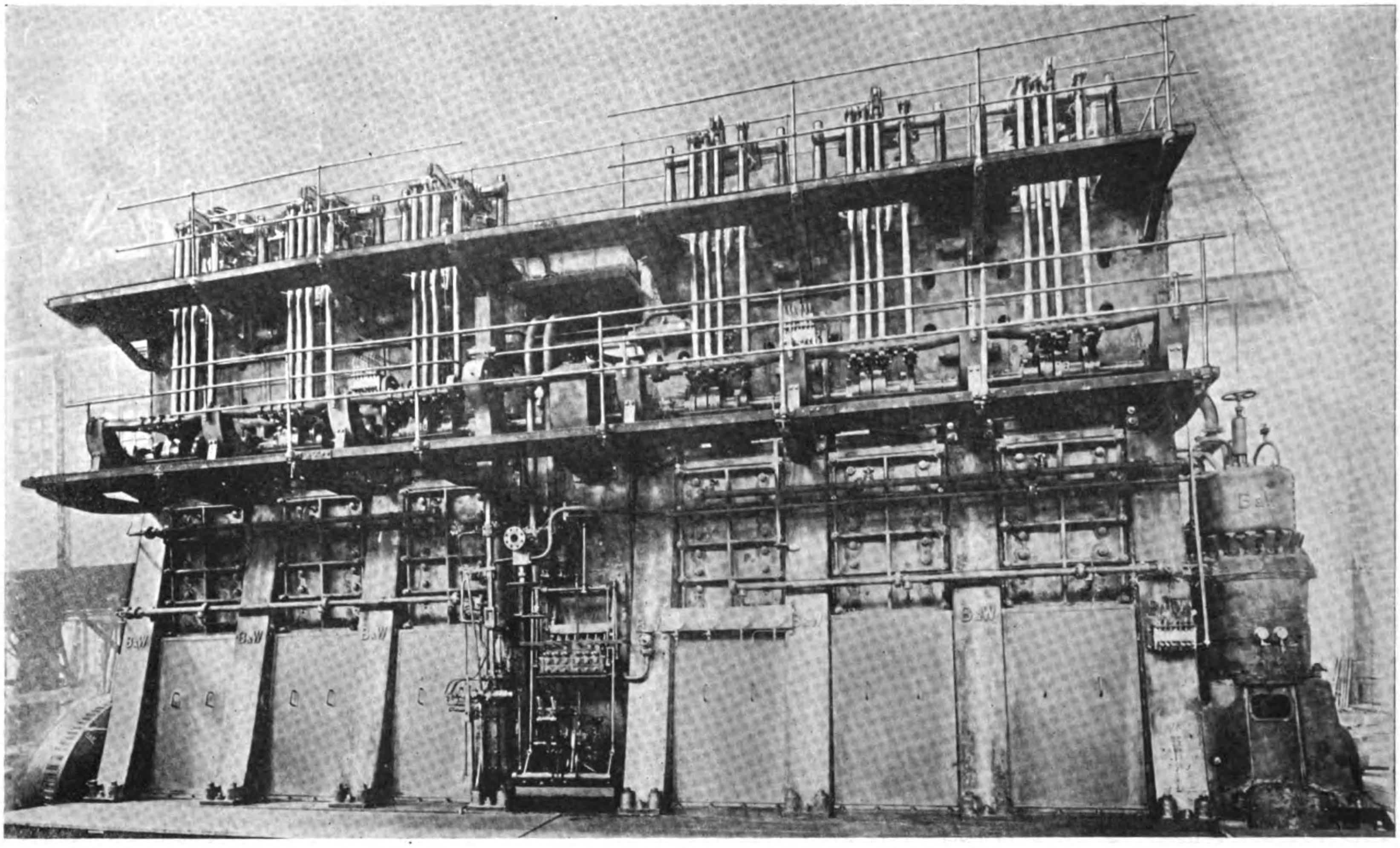
The United States Shipping Board will receive the first set of 4,000 I.H.P. Diesel motors, the largest units that Messrs. Burmeister and Wain have designed and are now constructing. The second pair of these engines will be installed in the largest motorship in the world; viz, one of the 12,000-ton ships building for the East Asiatic Co. of Copenhagen. The first pair have already been erected and have completed the shop tests at the B. & W. works in Copenhagen. They will become part of the return cargo of one of the American provision ships now unloading in the Danish capitol.

The engines will weigh 700 tons and will probably be shipped either to Baltimore or Philadelphia. The photograph on this page is the first illustration of this engine published in this country. The bore and stroke are 740 m/m x 1150 m/m (29.11 in. x 45.27 in.) respectively and the revolutions per min. are 115.

"The Americans will,' said Mr. Blache, "in their standard motorship program adhere to the 10,000 and 13,000-ton types and will install powerful engines in order to obtain high sea speeds. The desire for increased speed and tonnage is, of course, a question for the shipowners to decide as there are no technical or engineering difficulties to prevent their adoption. When there are sufficient quantities of goods to be transported the larger vessel is more economical. The question of increasing the speed is to be answered by calculating whether or not the increased manufacturing and running costs will be recovered in the greater number of voyages completed during the year by the speedier ship."

It is the experience of shipowners that the higher classes of freight which pay the higher rates are generally placed on the faster vessel and thus this type of cargo boat will more easily get a full cargo for each trip than would a slower boat. Again, if a limited number of passengers are accommodated, as has been the practice in the larger European motorships, they will prefer the faster vessel.

To indicate the growth of the modern motorships as to size and speed it will be remembered that the first motorship "Selandia" did not have a speed of more than 10½ knots. The newer ships such as the M.S. "Chile," "Peru," "George Washington," etc. have a speed of 12 knots. The 8,500-ton motorship "Glenapp" built during the war by Barclay, Curle & Co., Glasgow, was powered with twin 8-cylinder engines totalling 7,000 I.H.P. at 125 R.P.M. She has a sea speed of 14 to 15 knots. Her fuel consumption is one ton an hour and the cargo carrying capacity is 8,460 tons. In addition to this she could transport 1,700 troops. The next vessel to be launched at Copenhagen for the East Asiatic Co. will be a 13,000-ton motorship.



One of a pair of Burmeister and Wain Engines each of 2000 i.h.p. Purchased by U. S. Shipping Board; r.p.m. 115, weight 350 tons, fuel consumption not over 0.14 kg., 0.31 lbs. per i.h.p.; daily consumption not over 13.3 tons (2240 lbs.) per 24 hr. day, with 4000 i.h.p. from two engines. Bore 740 m.m. (29.11 inches) stroke 1150 mm. (45.27 inches).

The letters B. and W. on the engine columns are a little more than the height of a man's head above the floor. The fuel pump body can be seen between the centre columns and it is to be noted that all the pump plungers are operated by the same cross beam. The fuel pump operates at the same speed as the main pistons so that there are two strokes of the fuel plungers to every injection of fuel into the cylinders. Hand priming of the fuel line to the injection valves is by means of the small plungers shown in the lower part of the pump body.

The air starting and fuel control is by means of the two small levers shown on the vertical quadrants under the fuel pump. Each lever controls three cylinders. They are shown in the stop position. To the left of the operator as he stands facing these levers is the pneumatic control of the valve gear for ahead or astern setting. Complete and positive interlocking devices are provided.

Moving the levers away from the operator over the quadrant first admits air to the cylinders, then after a few revolutions the left hand lever is quickly moved to the extreme position. This throws the after three cylinders on to oil in full-load quantities. Immediately upon hearing the oil "firing" in these cylinders the other lever is thrown over and both are brought back over the quadrant to the setting giving the speed required.

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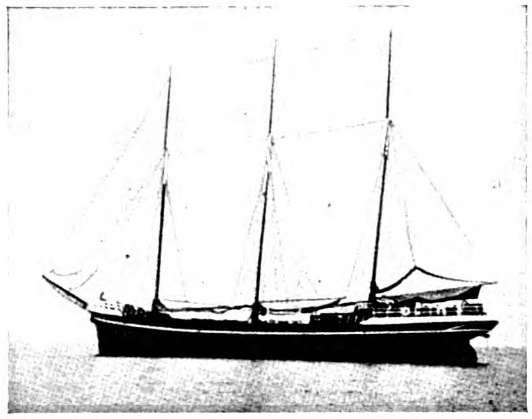
A Dutch Marine Oil-Engine

The Kromhout Hot-Plate Ignition Two-Cycle Type Engine

YEARS of operating service and manufacturing experience are the back-bone and foundation of many of the European designs of heavy-oil motors. It is the practice on the Continent and in Great Britain for concerns, when newly interested in any mechanical field, to follow along prior lines of success and derive all possible benefit from previous experience. The small and medium sized coasting and fishing vessels have always presented a most attractive market for a reliable, simple and economical engine, and a type of motor resulted which was particularly adopted for these classes of commercial crafts, by reason of their simple design and operation. In a way it is logical to expect that where stops are frequent and facilities always near at hand for repairing and overhauling, the personnel will let things slide and run to destruction without attention more frequently than when there is more of the element of safety dependent upon the continuous operation of the engine. This is witnessed by the regular routine generally laid out and followed concerning the overhauling and inspecting of running parts where the vessels make long voyages and regular runs.

A combination of design, materials and construction that can survive the test of neglect and abuse usually given to the engines of small coasters and work boats is an asset that any firm should consider invaluable. Evidence of the esteem and confidence placed in a product of this nature is given by the numerous successful boats powered with heavy-oil engines continuously ordered by shipowners in Holland, and sent on large ocean voyages as well as used for coastwise service. Quite a large number of small merchant motor-vessels have been fitted with the Kromhout marine oil-engine manufactured in Amsterdam at the Kromhout Works of D. Goedkoop, Jr., and also built in England by Plenty & Co. of Newbury, while at the present time the Dutch parent concern is seriously considering entering the American market.

As is usual with this type of engine, it works without inlet and exhaust valves, the admission of the air and its discharge of burnt gases taking place through ports in the cylinder wall, similar to American small two-cycle marine motor practice, which are opened and closed by the piston itself, and similar to the general run of surface-ignition oil-engines built in the U. S. A. Instead of the customary hollow bulb, ignition is by means of a cast-iron plate in the cylinder head, which is kept at the required temperature by the explosions, the fuel igniting on coming into contact with this plate. With some American engines a heated electric-coil is used for starting, but with the Kromhout designs the combustion-chamber is heated with a blow-lamp previous to starting, although this lamp is dispensed with as soon as the engine has turned over. The starting lamps, of course, are not required when the engine is running, even when running without load or at reduced revolutions for an extended period.



Auxiliary schooner L-164 ft.; B-27 ft. 3 in.; D-11 ft. 9 in. 3—90 h.p. Kromhout engines.

In the case of the smaller models, reversing is by mechanical gears, and this mechanism is so constructed that when going ahead the crank-shaft is directly coupled to the propeller-shaft by a friction-clutch. For running astern the motion of the crank-shaft is transferred in the reverse direction to the propeller-shaft by means of bevel gear-wheels and pinions which run in an oil-bath in a closed casing, the direction of running of the engine accordingly always being the same.

The fuel is fed to the cylinder by means of a plunger-pump, the stroke of which is regulated and varied by a centrifugal governor. The power of the engine is accordingly not governed on the hit-and-miss system, as in the case of some engines of this type, but by the quantity of fuel being regulated by changing the stroke of the pump's plunger.

In dealing with the supply of fuel to the suction side of the fuel-pump, where circumstances do not permit of the fuel-tank being higher than the delivery valve of the fuel-pump, it becomes necessary to lift or force the oil supply against a head. The Kromhout practice in this matter is as follows: There is provided on each engine a threaded hole with a blank plug near one of the air-valves in the base, and when the installation requires, this may be used to connect piping fitted with a non-return check-valve to the fuel tank. Thus the crank-case compression at each down stroke provides pressure of about 3½ to 4 lbs. per sq. in., which is transmitted to the fuel tank and which lifts the oil to the suction side of the fuel-pump.

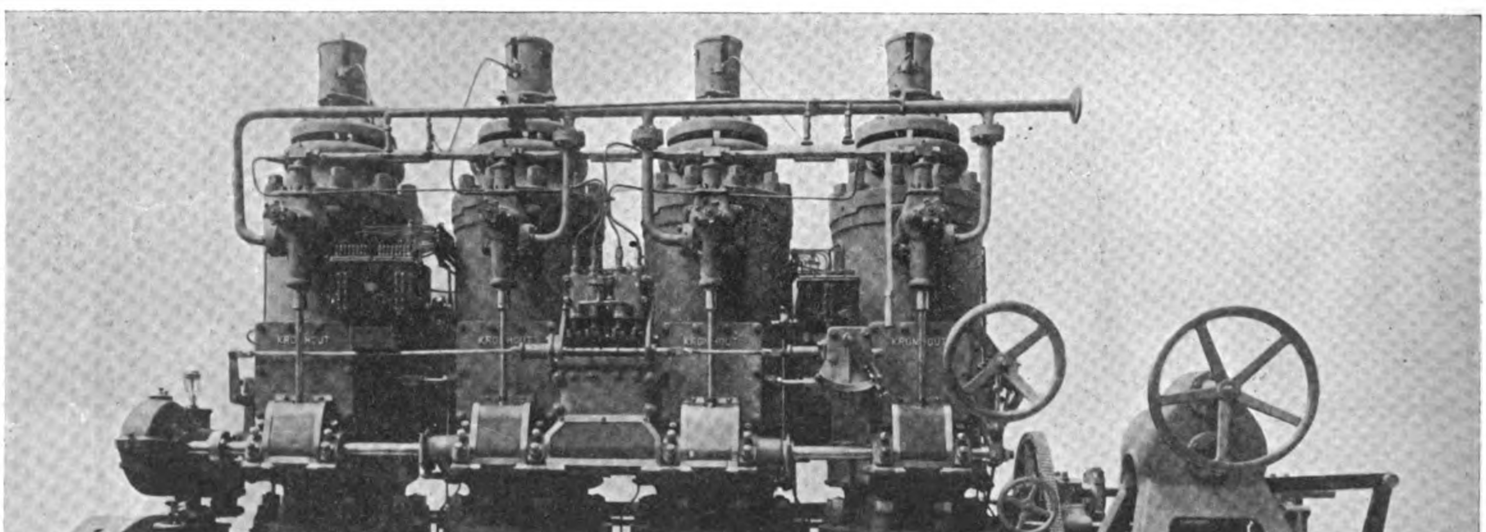
It may be remarked that though direct connection is thus established via of the inlet by-pass to the crank-case between the working cylinder, when exhausting, and the fuel tanks; the many years of actual use of this method indicate that the probability of crank-case explosions with possible damage to the fuel-tanks is quite beyond the range of accidental occurrence. We consider that it may be advisable to provide a duplicate check-valve on the pressure line in case the valve should stick or fail to function from any cause, but possibly the maker's experiences show the same to be unnecessary.

In the larger units starting-pressure may be obtained by storing compressed products of combustion at a suitable pressure by means of a check valve fitted as a relief connected to the combustion-chamber of each cylinder. This valve may be operated to charge the storage-tanks when and as desired, the pressure suggested being 200 lbs. per sq. in.

The general impression immediately gained upon inspection of the engines is the accessibility of all parts and the well-arranged controls for manouvering as may easily be seen from the view of the 180 h.p. four cylinder engine equipped with air-starting and a reversing clutch. Starting from the forward or flywheel end we see the vertical governor-shaft gear driven off the main crank-shaft; an extension of this vertical shaft carrying a helical gear driving the cam-shaft running fore and aft on the engine, which operates the air-starting valve rods. The fuel-pump body is located between the centre pair of cylinders on the operating side of the engine and the hand control may be plainly seen mounted above the cam-shaft between cylinders Nos. 3 and 4. The automatic governor-control is by a bell-crank at the forward end of the engine attached to an extension of the hand control shaft. The operating gear for the fuel-pump plungers is concealed by a cover plate extending from the pump body down over the cam-shaft. Gang oilers for force-feed lubrication are mounted in the other two spaces between the end cylinders and within the direct vision of the operator.

Starting-air is supplied to each valve from a common pipe with flanged joints at each branch. The hand wheel for the gear-operated reverse clutch is within easy reach of the operator and seems to be of sufficient size for easy operation.

One of a pair of 275 h.p. engines installed in the tankship "Hera" owned by a subsidiary company of the Royal Dutch-Shell Co. is illustrated and shows the fuel-pump mounted directly over the centrifugal governor and directly connected to the same. Hand control of the engine speed in



180 h.p. Non-Reversible Kromhout engine showing control of Air Starting Fuel Pumps and Reversing Gear

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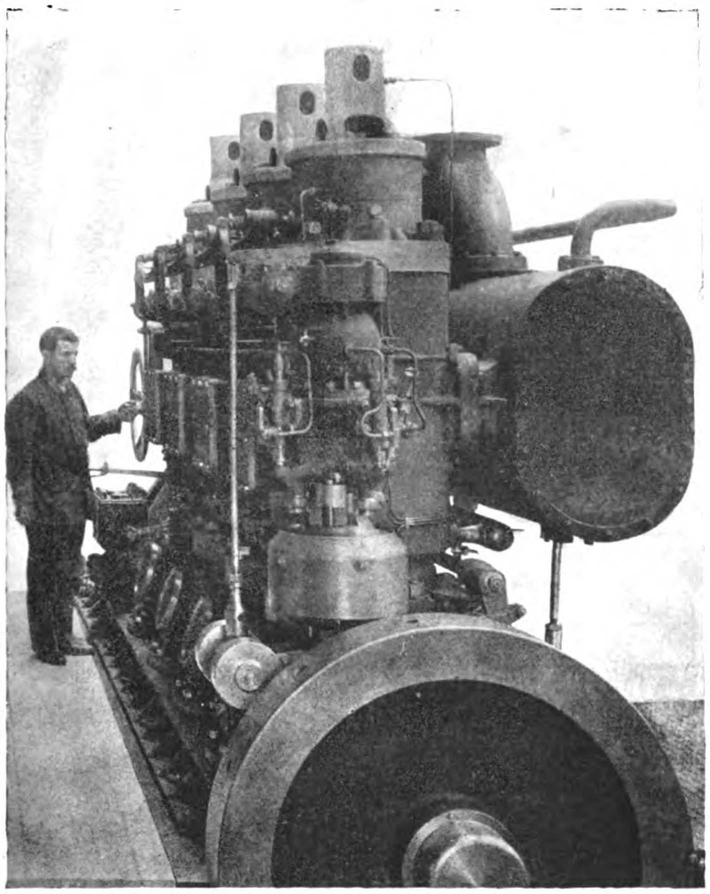
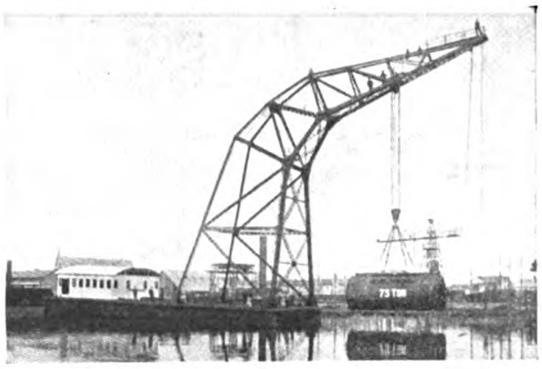


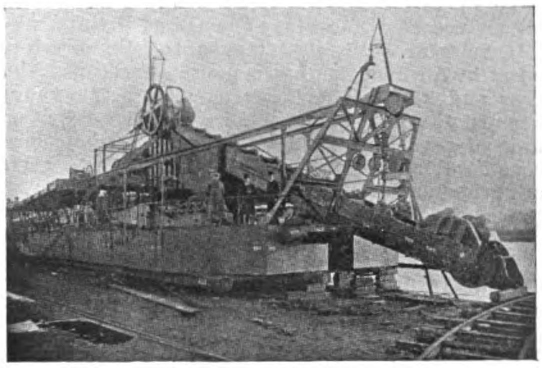
500 h.p. Direct Reversible Kromhout Marine Oil Engine. Showing centrifugal governor mounted on vertical shaft

this case is by means of a wheel at the after end connected by rods and levers to the under side of the governor. This gear actuates by mechanical means the governor control, thus varying the speed at which throttling of the fuel pump occurs. There is a thin diaphragm projecting vertically down into the centre of the combustion space and onto which the fuel is sprayed in a general horizontal direction. This thin diaphragm is first heated by conduction of heat from the heavy plate bolted over the top of the combustion-chamber which is heated by direct contact with a blow-torch firmly secured to the engine. It is said to be easily kept at the required temperature by the successive explosions even when running idle for long periods. In other designs of engines this problem is met by having a hot-bulb, hot-ball, hot-tube, or glowing electric-coil upon which the oil is sprayed.

The Kromhout oil-engine has undergone repeated changes in design and the policy of the company seems to be for continual progress toward general refinement along with sturdy construction. The dimensions of the cylinders of the 180 h.p. engine are 13 3/16 ins. bore by 13 13/16 ins. stroke, and the power is developed at a rated speed of 300 revolutions per minute. The slower and larger engine, whose cylinder dimensions are 15¾ in. bore and 17¾ in. stroke, runs at 230 revolutions, and it may be seen that their rated power is very conservative. The latter engine is direct-reversible.



Fitting-out crane powered with a 70 h.p. Kromhout engine



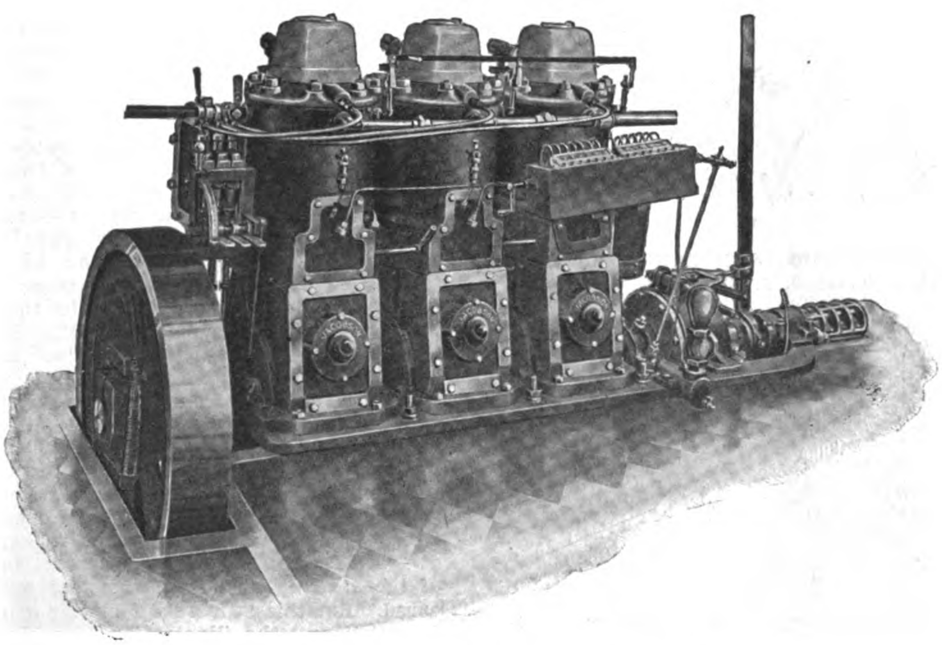
Dredge equipped with three Kromhout engines of 35 h.p. each

A Surface-Ignition Engine of Moderate Power

Notes Regarding a New American Marine Oil-Engine

THE general and widespread interest that has developed in the marine heavy-oil engine has brought to light a surprising number of engineering works which are equipped for, and experienced in, turning out marine power machinery in both high and low powers. This never fails to bring to mind the unanswered question—Why were the engine-building resources of the country entirely devoted to the production of steam-engines and boilers or their parts—when with equal experience brought to bear on quantity production of Diesel and surface ignition engines and their parts, the results would have been satisfactory and tangible; much as in the case of the propulsion machinery for the towing barges built for the New York State Canal, which are steam-equipped?

The few large parts found in the average Diesel-engine designed for 500 to 1,100 h.p. are not comparable with the weighty and bulky boilers and engine castings used in steam installations. Furthermore, one of the most valuable—because so readily available—assets of the country's manufacturing power has been completely overlooked in ignoring the small shop facilities for turning out at least some of duplicate parts for engines, also engine-room auxiliary sets, which, when completed, would not have had to wait upon the production of a set of boilers, and when installed would above all not have required coal for fuel. They could have been started right off with the oil-fuel obtainable at all times and would have really met the emergency for which the ships were intended.



A Jacobson surface-ignition marine oil-engine

In describing the Jacobson marine oil engine built by the Jacobson Gas Engine Co. of Albany, N. Y., we are dealing with a low powered motor of the surface-ignition type, which is built along conventional and generally accepted designs. On the machine illustrated is shown an extra large flywheel at the forward end for this size of engine, and for heavy-duty in work boats of all sorts this practice has many followers. At the flywheel end are the fuel-pumps mounted on the end cylinder where they are under the direct action of the governor, which can be seen built into the flywheel.

Easy access to the working parts of the fuel-pump body is claimed, and although we have not made a personal study of the engine, the illustration seems to bear out this contention and the fuel-pipes may be seen connected just under the hot-head device on each cylinder. The force-feed lubricator is driven off the crankshaft at the after end between the end cylinder and the reversing-gear. From the lubricator the oil is lead under pressure to the cylinder, main-bearings and connecting-rod ends and to the thrust-bearing located aft of the reversing-gear and clutch. All other working parts of the engine are lubricated by grease cups. The crankcase coverplates illustrated on the engine indicate by their size a generous amount of accessibility for an engine of this type.

No other attachments are provided in the cylinder head beside the fuel spray-valve and the hot-head device—exhause and scavenging inlet-valves being replaced by ports in the bottom of the cylinder uncovered by the piston at the bottom of its stroke. As may be presumed, the operation is carried out on the two-cycle principle with crankcase compression of air for scavenging and subsequent compression, which is carried up to about 200 pounds pressure per square inch.

At all speeds under the maximum, which is set by the governor, hand control is provided directly on the fuel-pumps for each cylinder by which means one-quarter of the rated engine speed may be obtained. At the moment of writing we are unable to give cylinder dimensions or revolutions of these engines, but understand that the design meets the usual requirements for this type of power installation in general work-boats, fishing vessels and engine-room auxiliaries.

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Proposed Method of Maneuvering Vessels

New Design of Reversing Rudder

THERE have been a considerable number of experiments conducted both in this country and in England, for the purpose of developing the design and construction of so called "reversible rudders." This term is meant to indicate that reversing is not accomplished by the change in direction of rotation of the propeller nor the feathering of the blades. The rudders are made of two similar halves, one controlled by or revolved about a solid rudder stock, and the other attached to a hollow shaft concentric with the inner stock. Reference may be had to the Patent Record in our July issue for rudder that has been developed in this country and to the illustration on this page of the Kitchen rudder referred to in the patent record as developed in Great Britain.

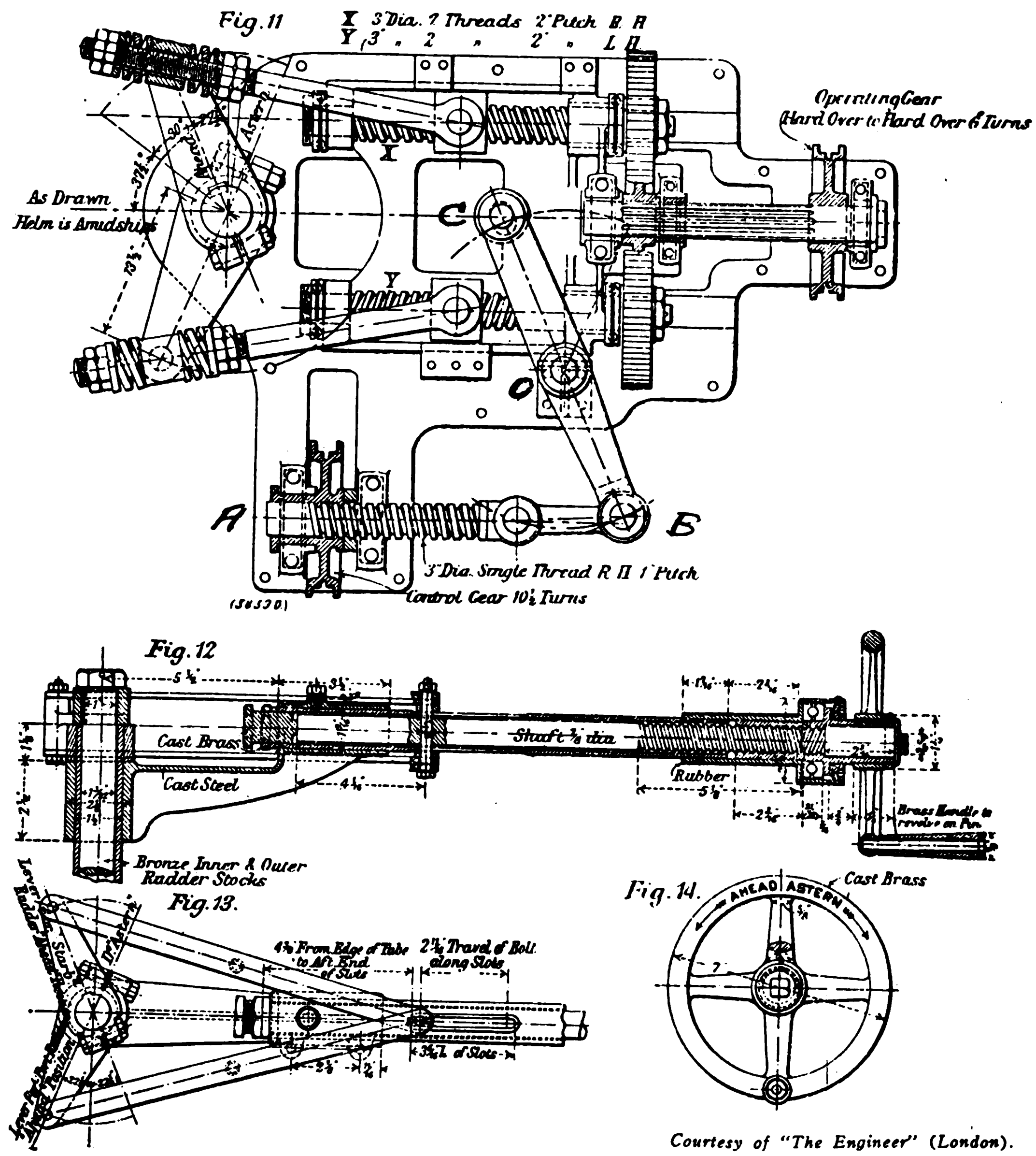
Particular interest attaches to the development in England in that a motor vessel of 1,100 tons and 950 B.H.P. is reported as being under construction, to which will be fitted the British type of reversible rudders. For small motorboat considerable success is indicated for this scheme of reversing for the small boats—the manouvering trials of a 50 ft. Admiralty Pinnance showed excellent control of the boat was possible.

The results claimed for any device of this nature are first—the simplification of the valve-gear of the propelling engine and the elimination of the necessity to provide a large storage capacity for manouvering-air, for, after the initial start the engines would continue to run continuously in the same direction. Also it is suggested that all controls for manouvering the boat would be under the direct supervision of the pilot or helmsman.

The essential parts of the Kitchen rudders consist of two curved deflectors formed of parts of a circular cylinder, partly enclosing the propeller. They are shown in perspective in Fig. 1. Both deflectors are pivotted at the top and bottom on common centres. One of the deflectors is operated by a solid shaft "A" and the other by a hollow shaft "B" concentric with the solid shaft. By suitable mechanism the deflectors or rudders are made to turn together in the same direction or equally in opposite directions. Some of the possible positions are shown in Figs. 2 to 8.

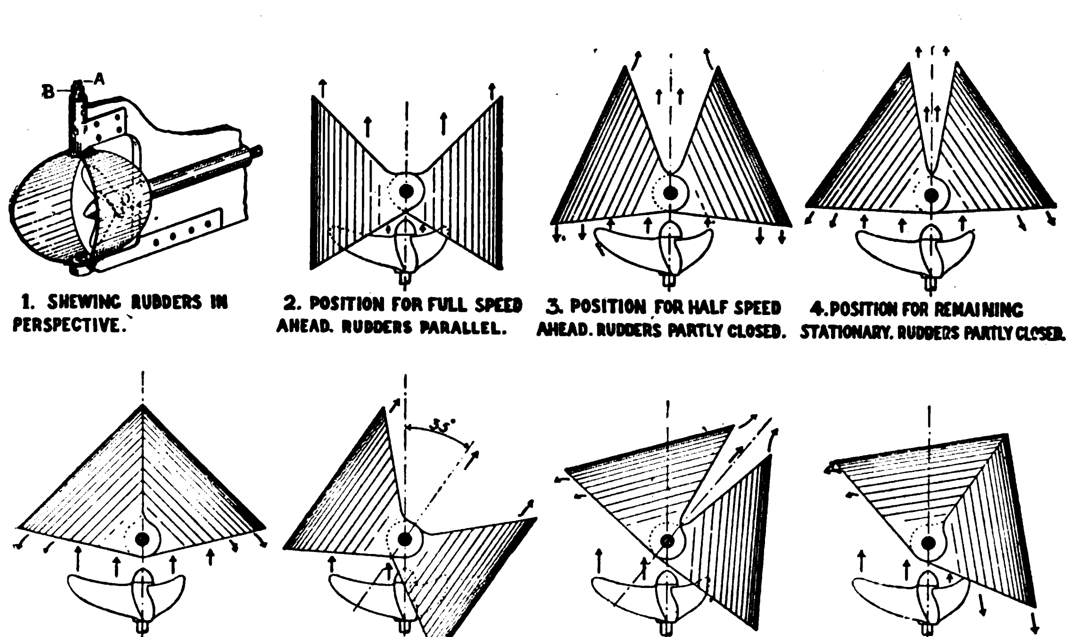
Simplification of the control gears for turning and backing is of course highly desirable and for boats which may be manouvered by hand the tiller shown in Figs. 12 and 13 is a neat arrangement. The whole tiller can be swung about the centre of the rudder stocks as usual for turning. For slackening speed and backing the hand wheel, Fig. 12 and 14, is turned. It is claimed that the neutral position, dead-stop, for any installation of the rudders is the same for all engine speeds, if the revolutions of the same should change. Then turning the hand wheel in one direction from this position will further close the rudders and the boat will gain sternway and turning the wheel in the opposite direction will open up the rudders and the boat will advance. In operating the hand-wheel with any amount of "helm" the internal shaft threaded at the forward end and carrying a gudgeon-pin at the after end advances or recedes in the threaded sleeve turned by the wheel. Links connect the gudgeon-pin to the tiller arms keyed to the inner and outer rudder stocks, thus opening or closing the curved rudder-blades and controlling the speed of advance or astern motion.

Fig. 2 shows a plan view of the steering-gear applicable to the larger sizes and which can be operated from a distance by ropes or chains running over pulley wheels. The tiller arms are connected to links as before but here supplied with buffer springs to absorb the impact stresses which may be caused by rough water pounding against the rudders. The links lead to nuts travelling on left and right hand screws which are turned through gearing by the wheel as shown. This part of the gear opens and closes the rudders simultaneously and by mounting the entire mechanism on a carriage it can be swung about the rudder stocks by the threaded link and rocker arm shown at ABC. The hub of the control pulley has an internal thread engaging the thread on the link—thus giving the motion required. The illustration shows the gear in the following condition—the rudders should be half closed with the links in midposition on the right and left hand screws and from the position of the helm control screw the carriage should be shown in a hard over position. However, if the fulcrum of the rocker (BC) is at O we do not see how the correct functioning of the *steering* portion of the gear would be secured. But this is merely a matter of detail construction and the illustration is perhaps drawn more for illustrating the principle rather than as a working design.



Courtesy of "The Engineer" (London).

Types of Operating Gear designed for the Kitchen Reversible Rudders



1. SHEWING RUDDERS IN PERSPECTIVE. 2. POSITION FOR FULL SPEED AHEAD. RUDDERS PARALLEL. 3. POSITION FOR HALF SPEED AHEAD. RUDDERS PARTLY CLOSED. 4. POSITION FOR REMAINING STATIONARY. RUDDERS PARTLY CLOSED.

5. POSITION FOR FULL SPEED ASTERN. RUDDERS CLOSED. 6. POSITION FOR FULL SPEED AHEAD TO PORT (OR STARBOARD) 7. POSITION TO REVOLVE SHIP WITHOUT PROGRESSION. 8. POSITION FOR GOING ASTERN TO PORT (OR STARBOARD)

Diagram illustrating the operation of the Kitchen reversing rudder

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"Motorship" Illustrated Patent Record*

Selected Abstracts of Recent Published Patents of Internal Combustion Engines

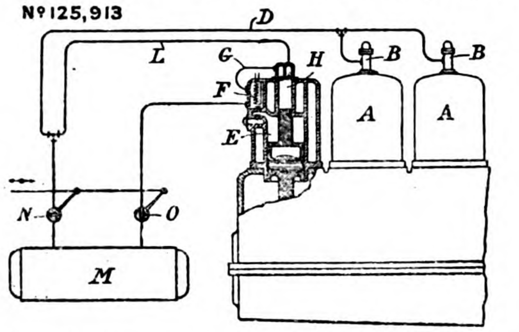
Copies of original specifications may be obtained for five cents each, by addressing the "Commissioner of Patents, Washington, D. C."

***Compiled and described by H. Schreck, Memb. Amer. Soc. Mech. Eng'rs**

125,913. December 20, 1918. Starting Diesel Engines. B. E. D. Kilburn, of London, England. A communication from Sulzer Bros., Switzerland (British Patent).

This invention relates to an arrangement for the purpose of quickly getting the high pressure injection air needed when an engine is started.

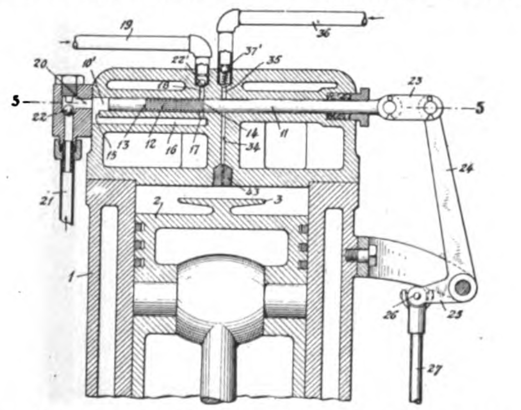
The illustration shows on top of the cylinders the fuel injection valves to which the injection air is supplied through the pipe (D). The intermediate pressure receiver of a multi-stage compressor (F) is connected to an air reservoir (M), which is also connected to the injection air line through the valve (N). When the engine is started, the valve (N) is closed, and



valve (O) is open, so that the intermediate stage receiver of the compressor is supplied with air from said reservoir. As soon as the air compressor by itself is able to maintain within the reservoir the necessary pressure for the normal working of the engine, the reservoir (M) is cut off from the receiver, and the valve (N) is at the same time slowly opened. In this manner the reservoir is again placed in communication with the injection air pipe, and the engine will then be working normally, the reservoir (M) serving to equalize the pressure of the air in the pipes (D) and (L).

1,290,374. Jan. 7, 1919. Oil feeding Device, J. C. Shaw, of Brooklyn, N. Y. Assignor to Keller Mechanical Engraving Co. of Brooklyn, N. Y.

This invention relates to a special fuel feeding and timing arrangement. A feed rod 11 or plunger is drawing the fuel through pipe 21 from the fuel tank 19'. On its suction the plunger will fill the fuel chamber 10' and through passages 16 and 17 fill also the parallel grooves 12 on the plunger. An air compressor 7 which runs at half engine speed will supply the injection air through pipe 36 and force at a proper time the oil collected in said grooves



through passage 34 into the cylinder against the hot plate 3 where it is broken up and vaporized.

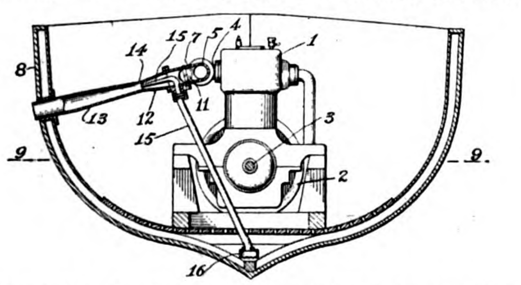
The push rod 27 which is operating said feed rod is actuated by a cam and by arrangement of asliding bar its stroke can be varied according to the load on the engine.

On the return or inward stroke of the feed rod, all air which had accumulated in the grooves will be pushed by the oil which remained in 10' and 16 through passage 18 and pipe 19 back into the oil supply tank'.

1,304,961. May 27, 1919. Boat-Bailing Means, J. Good, of Brooklyn, N. Y.

This invention refers to a contrivance on motor-driven boats, and consists in the employment of the exhaust gases for discharging bilge water.

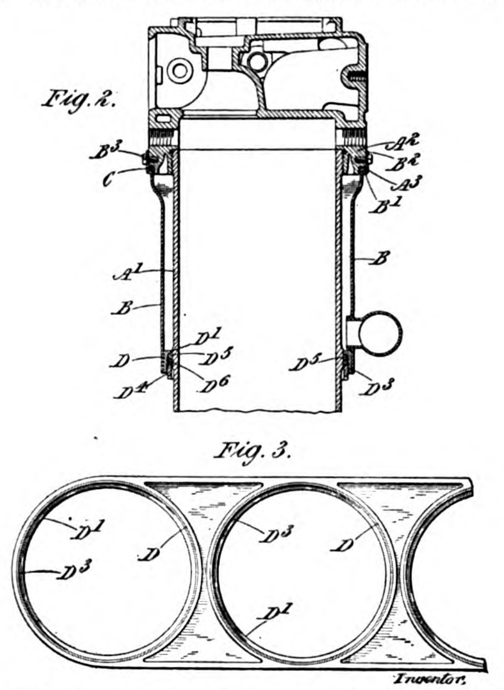
A supplemental exhaust pipe 7 is led from the main



exhaust line 5. The former contains a Venturi tube consisting of two tapering lengths of pipe 12 and 13, forming a throat or point of narrowest diameter at the point marked 14. The normal flow of the exhaust gases through this contraction thus provided results in a diminution of pressure at the throat 14, which is considerably less than atmospheric pressure and sufficient to raise a column of water several feet.

1,301,740. Apr. 22, 1919. Water Jacketed Cylinder. T. C. W. Pullinger, of Dumfries, Scotland, Assignor to W. Deardmore, of Glasgow, Scotland.

This invention refers to a light sheet metal water jacket. The cylinder carries at its upper end a shoulder which is screwed into the cylinder head or into

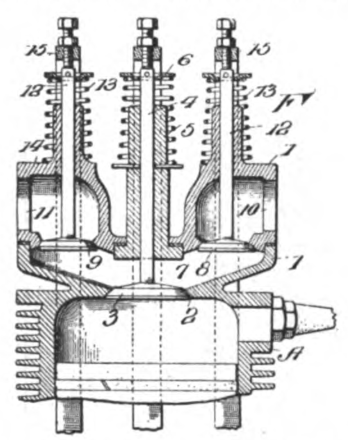


a block of cylinder heads. The jacket is screwed to this head and kept water-tight by a packing (C). Fig. 3 represents a gland plate which may be secured to the jacket by welding, and is made water-tight to the cylinder by packing (D^6) and the threaded ring nut (D^4).

1,304,735. May 27, 1919. Valve Mechanism. H. G. Blumberg, of San Antonio, Tex.

The invention refers to a special arrangement of the valves on a gas engine with the object of a better cooling of the exhaust valve.

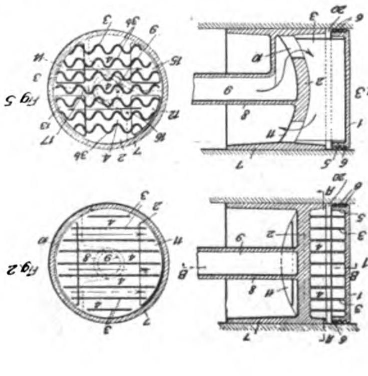
As is well known the exhaust valve requires very careful attention due to the heat to which this valve is permanently exposed, in the first place, by the explosion heat in the cylinder and, in the second place, by



the hot gases passing through the valve. As the illustration shows the inventor has improved this condition by using the valve on the combustion chamber for both, exhaust and inlet, by which means the inlet air will cool said valve on every second stroke. Exhaust and inlet are then finally controlled by the respective valves 8 and 9 which are arranged at a point distant from the high temperature of the combustion chamber.

1,292,882. Jan. 28, 1919. Piston, A. Riedler, of Berlin, Germany.

This invention relates to a novel design of a piston. The

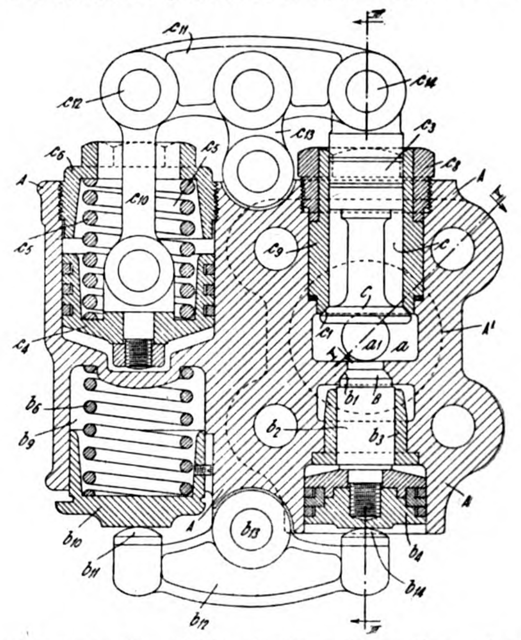


piston is arranged with a bottom and, spaced from this, with a false bottom which is integral with the piston body proper. The connecting webs between the piston bottom and the false bottom are designed to allow free expansion and contraction and are at the same time arranged so to form a proper conduit for cooling means.

1,305,193. May 27, 1919. Valve Fitting. G. J. Carter, of Bromborough, England

In the patent record of the Febr. issue of this year of our journal there was reported on a British patent, a similar patent has now been granted in this country on the same device to the same inventor. It will be of interest to the readers to see an additional detail of the construction of same.

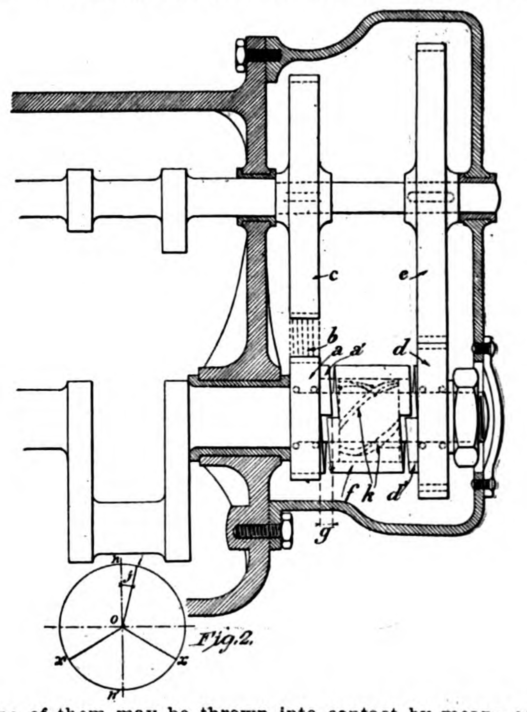
The patent in question is a valve casing which is



attached to the middle of the cylinder liner of an engine of the opposed piston type. The illustration shows a cut through said valve casing which fits with the flange A^1 against the liner. The casing carries the starting valve c, the safety valve B, in front (cut away) the injection valve, and a^1 is the opening to the cylinder. In order to make the valve arrangement as compact as possible the valve springs have been arranged sidewise instead of in an extension of the valves.

1,301,232. Apr. 22, 1919. Reversing of Internal Combustion Engines. H. Debauge, of Paris, France.

This invention refers to reversing of engines by changing the direction of rotation of the camshaft. A spur gear (e) and a sprocket wheel (c) are keyed to the camshaft, the former meshes with the pinion (d) and the latter drives through a chain (b) the pinion (a). Both pinions are loose on the crankshaft, and are revolving always in opposite directions. Either



one of them may be thrown into contact by means of their claws with the driving sleeve (f), which is keyed to the crankshaft by means of helicoidal keys (k). These keys are shaped in such a way that they take care of the alteration of the relative position of camshaft to crankshaft, that is, of the difference of the timing of the valves when the engine is to run one way or the other.

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